PB-222 871

VAPOR GENERATOR FEED PUMP FOR RANKINE CYCLE AUTOMOTIVE PROPULSION SYSTEM

LEAR MOTORS CORP.

PREPARED FOR
ENVIRONMENTAL PROTECTION AGENCY

DECEMBER 1972

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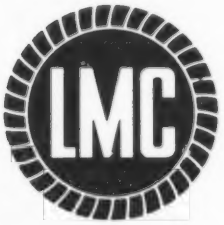
U.S. ENVIRONMENTAL PROTECTION AGENCY

Office of Air and Water Programs
Mobile Source Pollution Control Program
Ann Arbor, Michigan 48105



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16. Abstracts. The results are presented of preliminary design studies conducted to define feed pumps that satisfy the performance requirements for the following Rankine cycle automotive power systems which under development: (1) Steam Engine Systems; (2) Aerojet Liquid Rocket Company; and (3) Thermo Electron Corporation. The approach followed in this study consisted of establishing feedpump requirements for the three applications. The technical requirements as established by the contract are presented in detail. An additional requirement, which was not fulfilled, was to select a single design which satisfied both organic systems. Further investigations revealed that the feedpump developed by Lear Motors Corporation satisfies the basic requirements of the three system contractors. This feed pump was used as a baseline for the preliminary design and performance study presented here. A description of the Lear feedpump and its operating principle is presented. A summary of the flow requirements and estimated brake horsepower for the 3 system contractor feedpumps is shown graphically. Details of the contracts technical requirements and the Lear Motors approach for each system contractor are also presented.

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CONTRACT NUMBER 68-01-0437

FINAL REPORT

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Office of Air and Water Programs Publication Number APTD - 1358

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INTRODUCTION

This report is submitted in fulfillment of Environmental Protection Agency Contract No. 68-01-0437. The data presented herein are the results of preliminary design studies conducted to define feedpumps that satisfy the performance requirements for Rankine Cycle automotive power systems under development for the Environmental Protection Agency.

SUMMARY OF SYSTEMS UNDER DEVELOPMENT

System Contractor	Working Fluid	Type of Expander
Steam Engine Systems (SES)	Demineralized water.	Reciprocating expander.
Aerojet Liquid Rocket Company (ALRC)	AEF-78 (Organic fluid)	Turbine expander.
Thermo Electron Corporation (TECO)	Fluorinal 85 (Organic fluid)	Reciprocating expander.

II SUMMARY

The approach followed in this study consisted of establishing feedpump requirements for three (3) separate applications. This was accomplished in conjunction with the EPA and the three system contractors. The technical requirements as established by the contract and subsequent coordination meetings with the EPA and system contractors are summarized in Table I, and detailed in Section IV. An additional requirement was to select a single design which satisfied both organic systems. However, the final selection could not compromise significant gains in performance or cost reductions resulting from separate designs.

Further investigations revealed that the feedpump developed by Lear Motors Corporation satisfies the basic requirements of the three system contractors. This Lear feedpump was used as a baseline for the preliminary design and performance study presented in this report. A description of the Lear feedpump and its operating principle is presented in Section III-A. From the discussion presented in Section III-A, it can be seen that this Lear Motors pump has been designed for, and satisfies the basic requirements of; hermetic sealing of the working fluid, contamination of the working fluid, variable displacement capability, and low N.P.S.H. capability.

A summary of the flow requirements and estimated brake horsepower for the three system contractor feedpumps is shown graphically in Figure 1. These data which show system mass flow and brake horsepower versus percent of expander speed illustrate the relative magnitude of the respective requirements. Figure 2 shows the estimated feedpump performance, with a typical system operating envelope superimposed. These data show the affect of pump displacement-speed relationship on pump mechanical efficiency.

Details of the contracts technical requirements and the Lear Motors approach for each system contractor are presented in Section IV. These data, showing the estimated pump performance, estimated pump inlet envelope, and pump outline drawing were derived from actual test results and physical relationships of the Lear Motors 4th generation design feedpump.

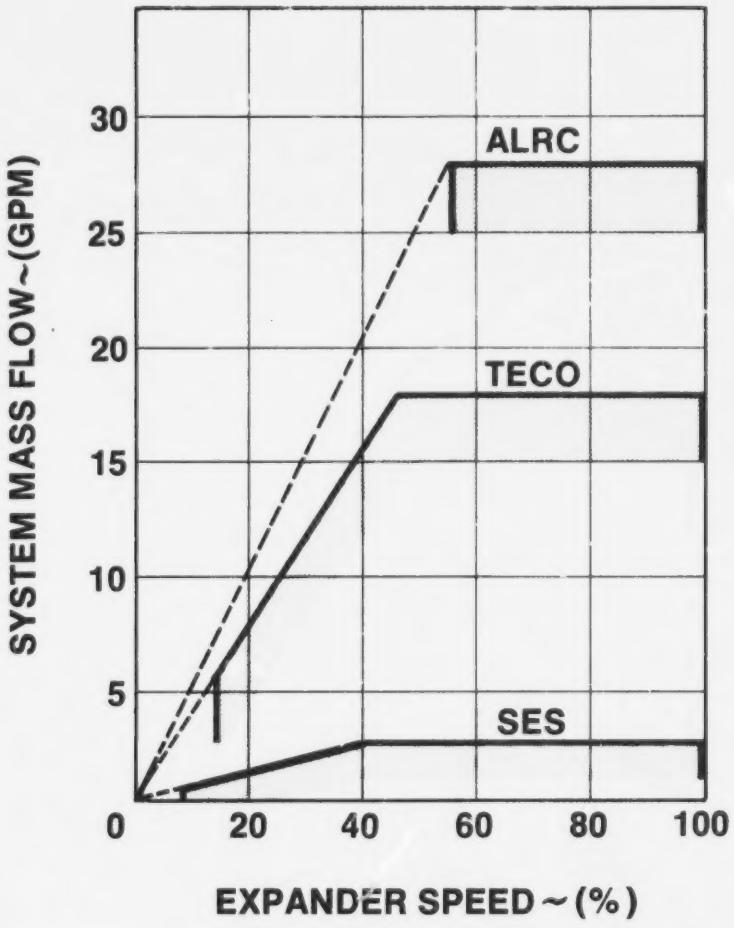
SUMMARY OF CONTRACT TECHNICAL REQUIREMENTS

	SES	ALRC	TECO
Pump Performance			
Type of Expander	Reciprocating	Turbine	Reciprocating
Expander Speed	300-2500 RPM	16,800-31,200 1.85:1 Speed Ratio	300-1800 RPM
Flow Range and Corner Condition	0-2.5 GPM Max system flow @ 40% speed	0.5-28 GPM Max system flow @ 54% speed	0-17 GPM Max system flow @ 45% speed
Discharge Pressure	1200 PSIA max.	1100 PSIA max.	1000 PSIA max.
Mechanical Efficiency	70% from 30-80% of flow max. 50% min. at 10% of flow max.	70% from 30-80% of max flow 50% min. at 10% of flow max.	70% from 30-80% of max flow 50% min. at 10% of flow max.
Pump Inlet Envelope			
Inlet Pressure	0.09 PSIA @ Start 7-50 PSIA operation	1.0 PSIA @ Start 10.0-40.0 PSIA operation	0.5 @ Start 5-90 PSIA during operation
NPSH Capability	Minimized to operate without boost	7 inches min. avail (.45 PSI @ 72 °F)	10 inches min. avail. (149 PSI @ 72 °F)
Temperature Range	180-250 °F Normal 32-275 °F Start	160-250 °F Normal -40-275 °F Start	160-250 °F Normal -40-275 °F Start
Temperature Transients	3° per second for 10 seconds		
Boost Pump	Only if needed	Only if needed	Only if needed
Pump Drive			
Drive Mechanism & Flow Range Control	Mechanical with provision for variable speed or variable capacity.	Mechanical with provision for variable speed or variable capacity.	Mechanical with provision for variable speed or variable capacity.
Working Fluid	Water - Demineralized (Pump to have provision for draining)	Organic S.G. = 1.793 @ 72 °F	Organic S.G. = 1.368 @ 72 °F

TABLE 1

ESTIMATE OF SYSTEM CONTRACTORS FEEDPUMP PERFORMANCE

SYSTEM CONTRACTOR FLOW REQUIREMENTS



ESTIMATED FEEDPUMP BRAKE HORSEPOWER

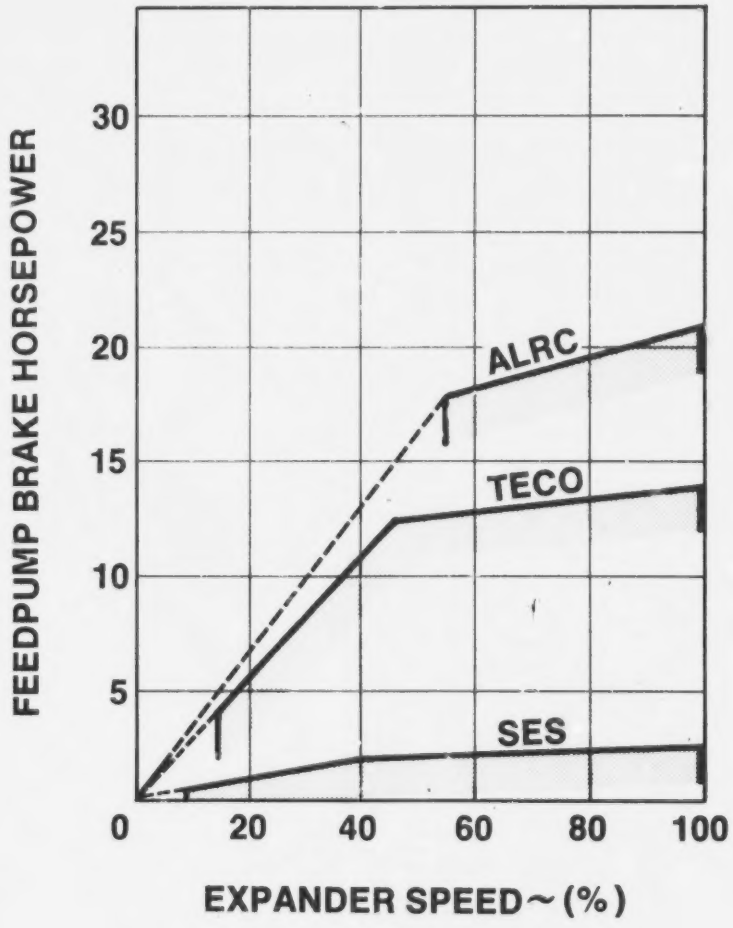
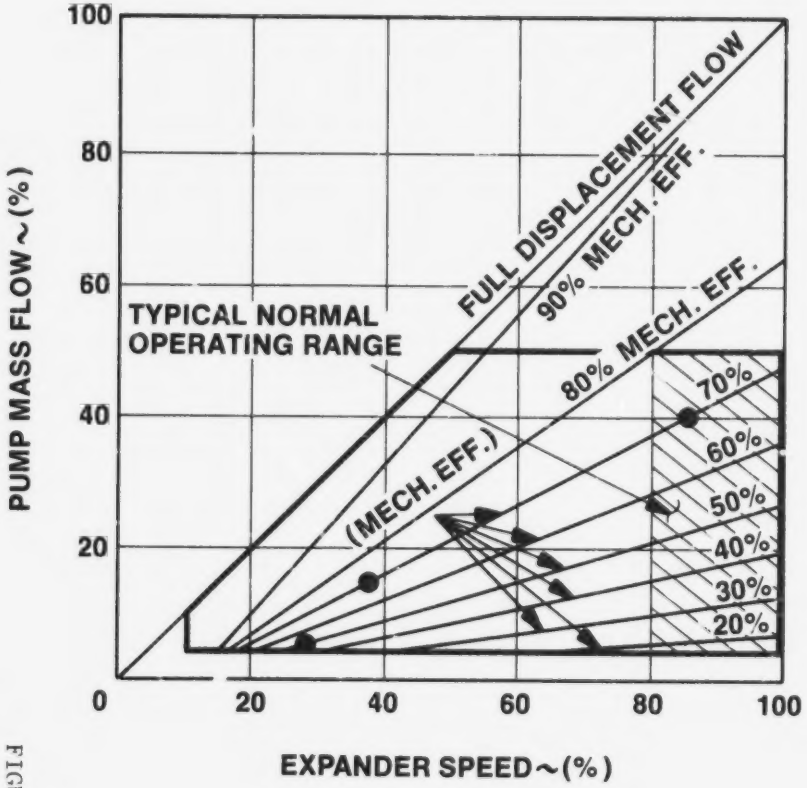
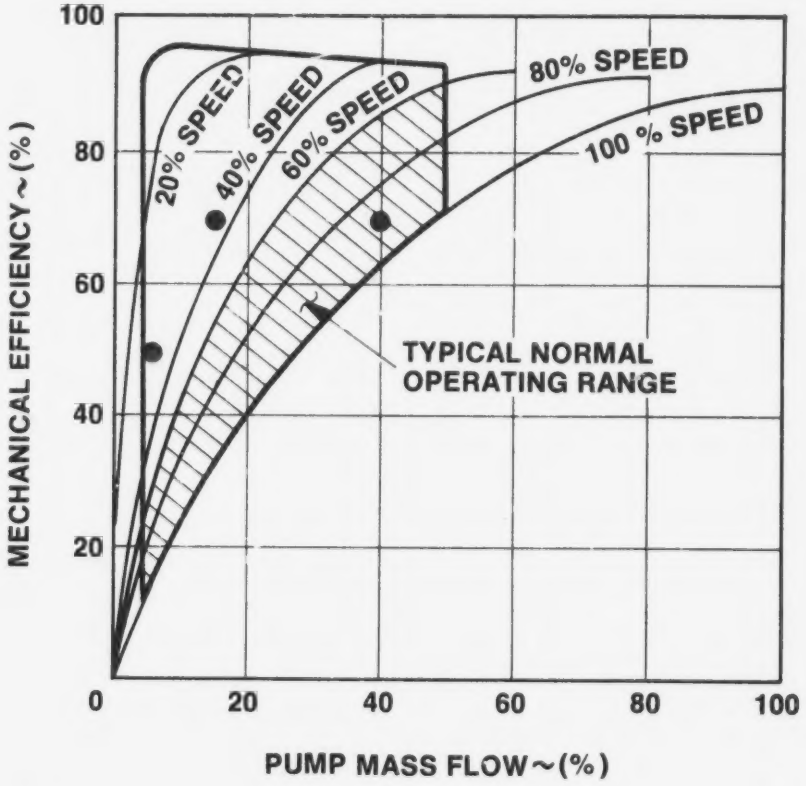


FIGURE 1

ESTIMATED FEEDPUMP PERFORMANCE





● CONTRACT MECHANICAL EFFICIENCY DESIGN POINTS

FIGURE 2

III TECHNICAL DISCUSSION

In Rankine cycle automotive power systems the feedpump must be capable of efficiently pumping the working fluid up to system pressure while operating at very low net positive suction head conditions and very high fluid inlet temperature to achieve reasonable cycle efficiencies. Three years ago, Lear Motors initiated the development of pump specifically designed for Rankine cycle applications. The results of this development have been used as the basis for these studies.

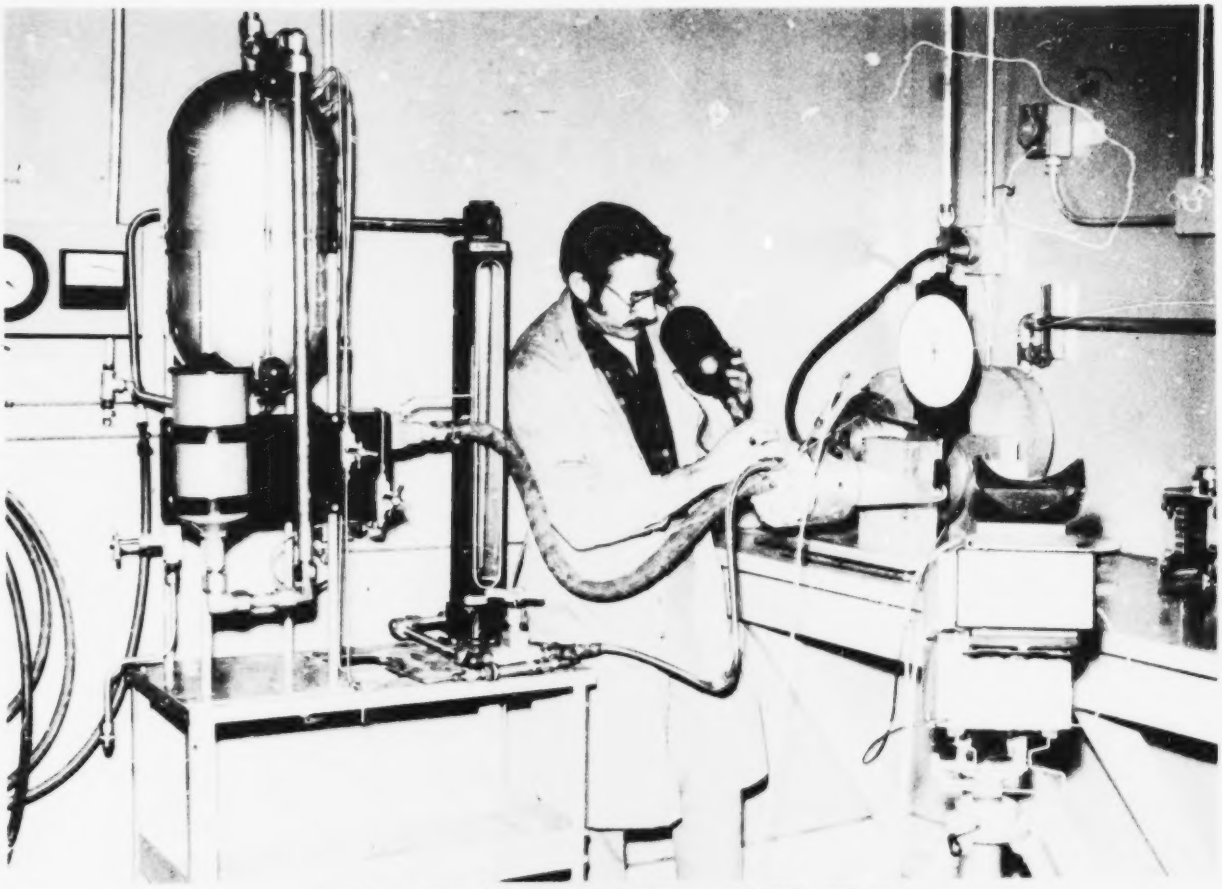
A. LEAR MOTORS FEEDPUMP OPERATING PRINCIPLE AND DESCRIPTION

The Lear Motors high pressure feedpump design, incorporates pistons reciprocating within close fitting cylinders filled with oil. A flexible diaphragm is positioned between each cylinder cavity and the working fluid pump chamber. The alternate suction and displacement of oil by the pistons actuates the diaphragms and thus results in an equal displacement in the working fluid chamber. No mechanical connection is required between diaphragm and piston.

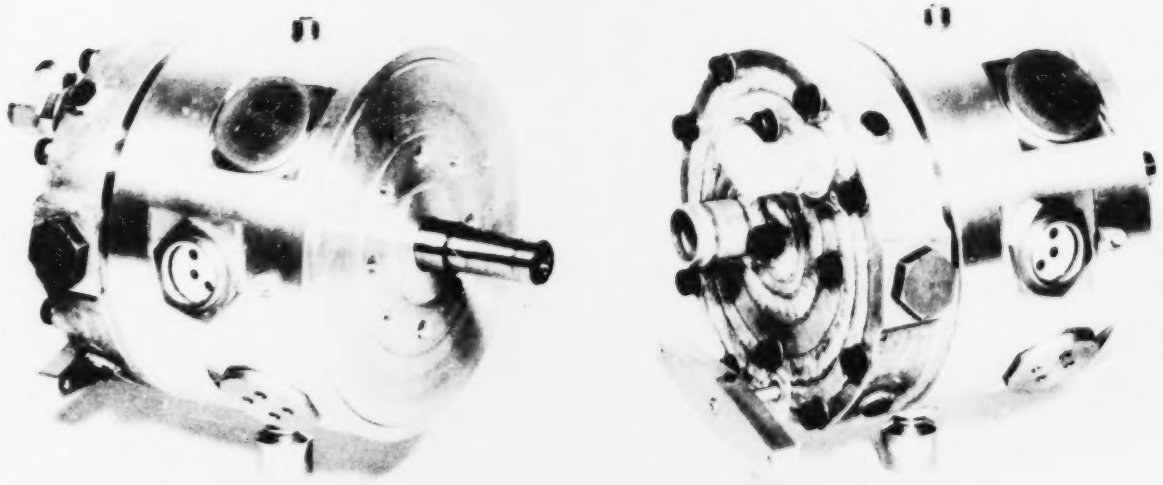
Since incompressable fluids are present on both sides of the diaphragm, pressure across the diaphragm is always equal. The pump may be operated at any pressure within the structual limits of the pump without stress upon the diaphragms or any reduction in diaphragm life.

The pump incorporates a fixed stroke, needle bearing, eccentric on a drive shaft which is supported by two ball bearings. The eccentric converts shaft rotary motion to reciprocating motion and is used to drive the pistons which are equally spaced around the drive shaft. On the displacement, or pressure stroke, the eccentric drives the piston up, pressurizing the oil chamber above the piston which actuates the diaphragm and displaces the working fluid in the pump chamber through the outlet valve. During the suction, or return stroke, the piston is held to the eccentric by a positive return band and as the piston, oil and diaphragm are drawn back, working fluid enters the pump chamber through the inlet valve.

The entire drive shaft and piston assembly is located within the pump's temperature compensating oil reservoir and is completely lubricated. The oil filled chamber above the piston is connected to the oil reservoir through a slot in the piston wall and a



LEAR MOTORS TYPICAL TEST STAND



LEAR MOTORS FEEDPUMP

FIGURE 3

triangular port in the side of the cylinder. The piston slot must be closed before oil can be displaced in the chamber above the piston. That portion of fixed mechanical stroke during which the piston slot is closed is the effective stroke of the pump. By varying the cylinder port with respect to the piston slot, the pump's effective stroke or displacement can be controlled. Each cylinder is linked mechanically, inside the pump, to an external pump displacement control shaft which can be actuated directly or with a small servo mounted outside the pump.

There are no sliding or rotating seals in contact with the pumped fluid, allowing the pumped fluid to be hermetically sealed from atmosphere and lubricating oil. All of the highly stressed moving parts in the pump are oil lubricated allowing conventional, low cost materials to be used throughout its construction. Efficient operation of the pump, during low N.P.S.H. conditions of Rankine cycle system, is made possible by utilization of large passages and valves designed for minimum flow losses throughout the inlet circuit of the feedpump.

This feedpump, designed specifically for use with the Rankine cycle system, is the result of a continuing feedpump development program at Lear Motors Corporation.

B. FEEDPUMP DESIGN STUDY

Pump size is determined primarily by the system flow requirements as defined by the high flow-low speed corner condition. This point and the mass flow at idle define the mass flow-speed characteristics slope. By extending the line to 100% expander speed the flow capability of the pump is defined. Since the system can't utilize the total capability of the pump, variable displacement or capacity is required to allow operation within the system envelope.

The corner condition of the pump performance map is determined by vehicle performance in terms of lugging and acceleration capability. If system performance is allowed to degrade then the slope of the pump characteristics can be changed. Thus, a compromise in system performance is required if one pump design is used for both organic fluid system contractors. For this study a penalty in system performance was considered undesirable. Therefore, a separate pump design was prepared for each system contractor.

After the high flow-low expander speed corner condition for each system contractor was established, the maximum speed capability of each pump was investigated based on an analysis of both radial and axial piston pump designs.

From this analysis it was determined that maximum speed capability of the radial pump is limited by the valve dynamics and speed at which the oil in the pump crank case may be sheared without excessive friction losses, rather than the dynamic unbalance of the fixed eccentric on the crank shaft. Further study revealed that the radial piston design would be physically smaller, lighter in weight, and less expensive to produce.

Utilizing the established maximum pump speed in conjunction with the system contractor flow and maximum pressure ripple requirements, the bore/stroke relationship and number of cylinders were selected. To keep the instantaneous total flow of the pump, for any given crank shaft angle, as smooth as possible, an odd number of cylinders was used. The number of cylinders and the bore/stroke relationship determine the displacement of the pump as well as its smoothness. Obviously the greater the number of pistons for a given flow the smoother the pump will be, however, pump cost and physical size must be considered. After close examination of the parameters involved and based on our four years of feedpump experience, the bore/stroke relationship and number of pistons which would best meet the individual system contractor feedpump requirements was determined.

The diaphragm seals for each pump were then examined. These diaphragms are the key to the hermetic sealing capability of the Lear Motors feedpump design. They also enable the pump to produce high pressure working fluid without utilizing the fluid to lubricate highly stressed moving components such as piston seals, gear teeth, or vane tips. The diaphragm effective area selection was based on several factors: 1) The maximum volume it must displace during each piston stroke. 2) The physical design of the seal. 3) The material used to construct this flexible sealing member.

As the Lear Motors feedpump has demonstrated excellent performance with a flat circular sealing member using an elastomer material, our major effort was directed toward finding elastomers

which had the physical properties required and were compatible with the two organic fluids used by the system contractors.

A separate study was conducted and limited tests for compatibility were conducted by the organic fluid system contractors on candidate elastomer materials selected and supplied by Lear Motors Corporation. The results of this study indicated that several of the elastomers evaluated, were acceptable for use as sealing diaphragm material. The suggested procedure and test results are shown in Section III-D of this report.

An additional investigation into the possible use of a bellows type metal diaphragm was carried out. However, due to the low cycle life and high costs quoted by the venders and available design literature, the use of this type of seal was deemed not practical.

Having established a suitable elastomer material for each of the system contractors and selected the flat circular diaphragm design, the minimum effective area required for long diaphragm life was determined. It is apparent that for a given piston displacement the larger the effective or flex area of the sealing diaphragm, the longer the diaphragm life. However, the physical size of the pump, namely its outside diameter, will become proportionately larger. Therefore, the minimum area of diameter of the diaphragm becomes an important individual pump design feature. This sealing member is simply a flexible incompressible barrier which separates two other virtually incompressible fluids: the most important physical characteristic of the elastomer sealing or separating members used in these pumps is the flex life capability of the elastomer. Fortunately, this is what most elastomers are compounded for and the selection of materials is large even when limited by the Rankine cycle system fluid temperature extremes and material compatibility requirements.

The designed maximum movement or flex used by Lear Motors is 8 to 10 percent change. For each system contractor a specific diaphragm effective area was determined based on the maximum volume displaced per piston and the 8 to 10 percent flex value which has proved successful in the Lear Motors feedpump. It must be pointed out that during most of the vehicles normal operation the feedpump is

functioning at a reduced displacement and the diaphragms would be flexing less than design.

Having determined the individual system contractors pump, bore and stroke, number of cylinders, pump speed, and outside diameter required by the pitch circle of the individual diaphragm seals, a preliminary analysis of each pumps bearing requirements and operating mechanics was made. As one of the basic strong points of the Lear Motors feedpump design is that all of the highly stressed bearings and moving parts required by the feedpump are oil lubricated, conventional low cost bearings and materials may be used, making the analysis of the lubricated section of the pump straight forward. One area in this lubricated section required special attention. The pumps unique outlet flow or displacement control, described in Section III-A Feedpump Operating Principle and Description, must be designed to keep orifice and passage losses low to maintain high pump mechanical efficiency.

The next basic area which was examined during this design and performance study involved the inlet passages and inlet valve sizing. To reduce the net positive suction head required by the pump for efficient operation, special attention must be given the working fluid inlet of the pump. A description of the term net positive suction head (NPSH) is given in the appendix. Basically, a positive displacement pump is not known for its low NPSH characteristics; any entrained vapor in the pumped fluid will cause the pump performance to degrade rapidly. To minimize the NPSH required by the feedpump, the inlet velocity of the working fluid must be kept as slow as possible so that the friction losses are kept to a minimum. Also the inlet valve spring must be kept as light as possible and still allow the valve to operate at high pump speed. Large inlet passages and inlet valves which reduce fluid flow velocities were designed into each of the feedpumps. Maximum inlet velocities of 5 to 6 feet per second were used in the design of the Lear Motors feedpump and each of the system contractor feedpumps. The estimated inlet envelope showing the NPSH required for each of the pumps is detailed graphically in Section IV of this report. These data, corrected for individual organic fluid specific gravity variations, demonstrate what each of the pumps would be capable of based on actual Lear Motors test data shown in

Section III-C of this report.

Further examination of the NPSH required by these feedpumps, although low for positive displacement pumps, indicates that some amount of sub-cooling or boost pressure is required to meet the system contractor requirements. Sub-cooling of the working fluid results in a significant loss in Rankine cycle system efficiency due to the power expended in the cooling of the fluid as well as the heat required to bring it back up to a temperature after it has been pumped. Adding boost pressure or head to the inlet of the pump by placing the working fluid reservoir physcially higher than the feedpump is not practical in an automotive Rankine cycle system due to the limits of the engine compartment and the added expander back pressure. However, increasing the inlet pressure to the feedpump with an independently driven centrifugal pump demonstrates several advantages; the centrifugal type of pump has inherently low NPSH characteristics and can be designed to handle significant amounts of vapor. A centrifugal pump can operate at and below the systems working fluid vapor pressure line, allowing greater cycle efficiencies through the minimum sub-cooling required. By mounting a small separately driven centrifugal pump low in the system the feedpump may be conveniently mounted and driven without regard to the systems liquid level. The vehicle will be operational with the absolute minimum sub-cooling because the independently driven centrifugal pump will provide some liquid to the feedpump regardless of the expander speed. An independently driven centrifugal boost pump will allow the system to be started after a hot shutdown. It was concluded after examining the parameters involved that a small independently driven centrifugal pump is important to the efficient operation of an automotive Rankine cycle system.

Having concluded the preliminary design portion of this study, an outline or installation drawing for each pump, shown in Section IV, was drawn. It must be pointed out that these pumps were designed specifically to meet the individual system contractors requirements and that their physical size is determined primarily by the systems high flow-low speed corner condition not just the maximum system flow required.

Example: A positive displacement pump with flow-speed

SYSTEM CONTRACTOR FEEDPUMP

DESIGN SPECIFICATIONS

	SES	ALRC	TECO
Configuration	Radial Piston, Diaphragm seal	Radial Piston, Diaphragm seal	Radial Piston, Diaphragm seal
Number of Cylinders	5	7	7
Bore	.68 inches	1.25 inches	1.18 inches
Effective Stroke	.41 inches	.75 inches	.67 inches
Diaphragm seal effective are per cylinder	2.08 in^2	6.51 in^2	5.43 in^2
Pump Design Theoretical Flow @ 85% Vol. Eff.	6.2 GPM @ 100 percent speed	52 GPM @ 100 percent speed	40 GPM @ 100 percent speed
System Flow Requirement	2.5 GPM @ 40 percent speed	28 GPM @ 54 percent speed	17 GPM @ 45 percent speed
Pump Speed (Max. Design)	2400 RPM	2000 RPM	2000 RPM
Pump Displacement Vol/Rev.	.77 in^3/Rev.	6.52 in^3/Rev.	4.64 in^3/Rev.
Inlet Velocity @ Max System Flow	5.1 FPS	5.9 FPS	5.5 FPS
Estimated Pump Weight with			
1. Iron Housing	16 1/2 pounds	108 pounds	66 pounds
2. Aluminum Housing*	12 pounds	81 pounds	46.5 pounds

* Additional stress analysis required.

TABLE 2

characteristics capable of producing 26 GPM (gallons per minute) at 50 percent or one-half speed is theoretically capable of producing 52 GPM at 100 percent or maximum speed. While the system may only require 26 GPM mass flow the pump must be physically large enough to produce 52 GPM. The only reduction in pump size, due to the 26 GPM system requirement, will be working fluid flow passage area. From this example it can be seen that any reduction of the system contractors high flow-low speed corner requirement will result in a significant reduction in the feedpump size and weight.

C. FEEDPUMP PERFORMANCE STUDIES

Estimates of the individual system contractor proposed feedpumps performance and inlet envelope are shown graphically in Section IV. These estimates were obtained by extrapolating actual Lear Motors Corporation feedpump test data. The procedure and feedpump rating criteria used in conducting the tests, which were run specifically for this study, are shown in the Appendix.

The proposed feedpump estimated performance curves, showing pump brake horsepower, mechanical efficiency, and flow vs. pump and expander speed, demonstrate the broad range over which these pumps can operate. The lower section of these pump performance maps, indicate the flow characteristics of the pump at various speeds and displacement levels. Superimposed are the 50 and 70% mechanical efficiency lines denoting how the pump performs with respect to contractor performance requirements. Also indicated is the pumps full displacement flow line the slope of which is determined by the individual system maximum flow-minimum expander speed corner condition and the volumetric efficiency of the pump. Based on Lear Motors Corporation feedpump data, full displacement volumetric efficiency of the proposed pumps will be 85 to 90% depending on outlet pressure and inlet N.P.S.H.

Feedpump inlet envelope estimates are also shown in Section IV, under the individual system contractor heading. These inlet envelopes, superimposed over the contractors working fluid vapor pressure characteristics, demonstrate the N.P.S.H. required by the pump to maintain efficient operation. The pressure and temperature limits during normal and extreme operating conditions, as outlined by the

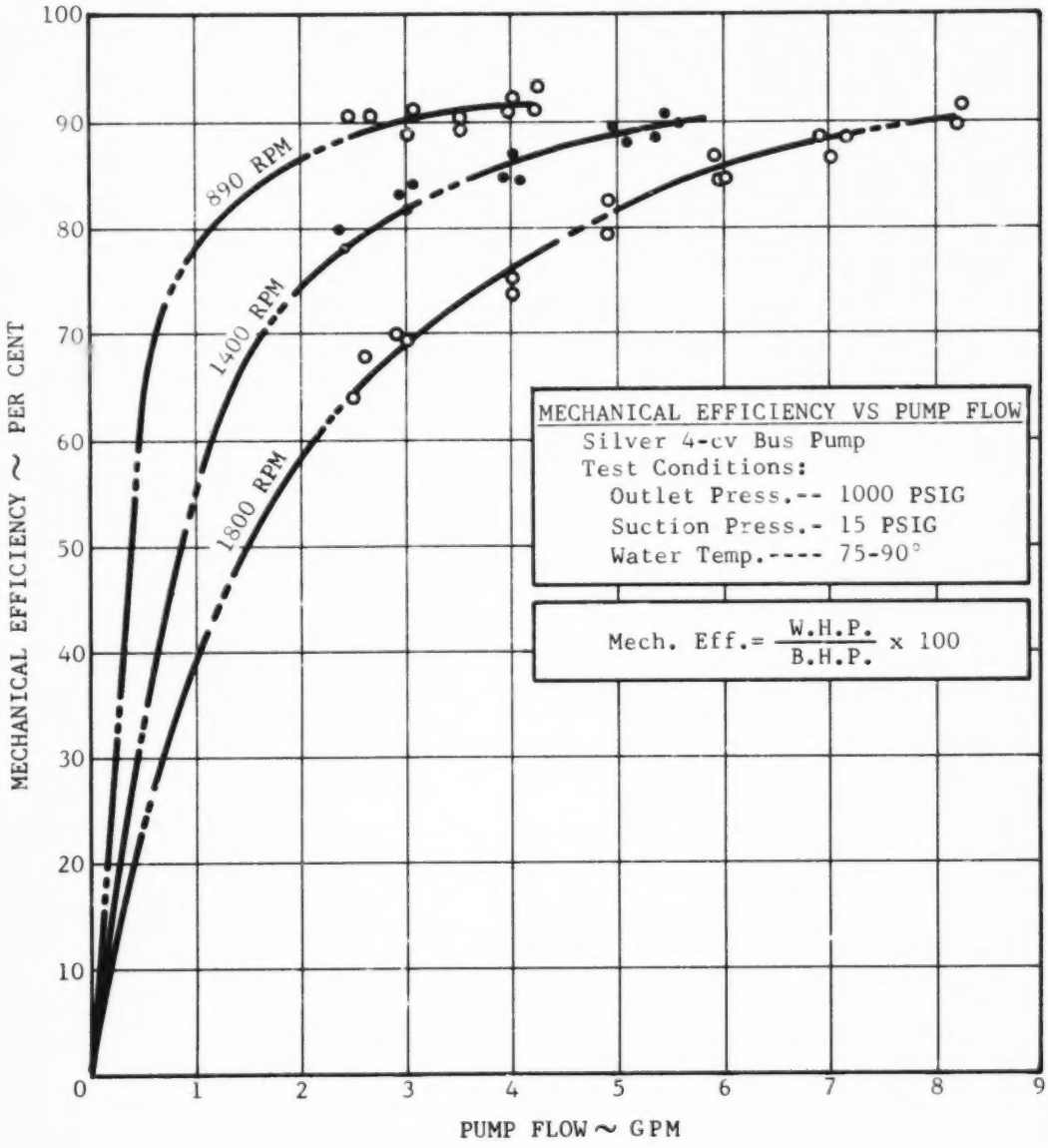

MECHANICAL EFFICIENCY ~ PER CENT
PUMP FLOW ~ GPM
890 RPM
1400 RPM
1800 RPM
MECHANICAL EFFICIENCY VS PUMP FLOW
Silver 4-cv Bus Pump
Test Conditions:
Outlet Press.-- 1000 PSIG
Suction Press.- 15 PSIG
Water Temp.---- 75-90°
Mech. Eff.= W.H.P. / B.H.P. x 100

FIGURE 4

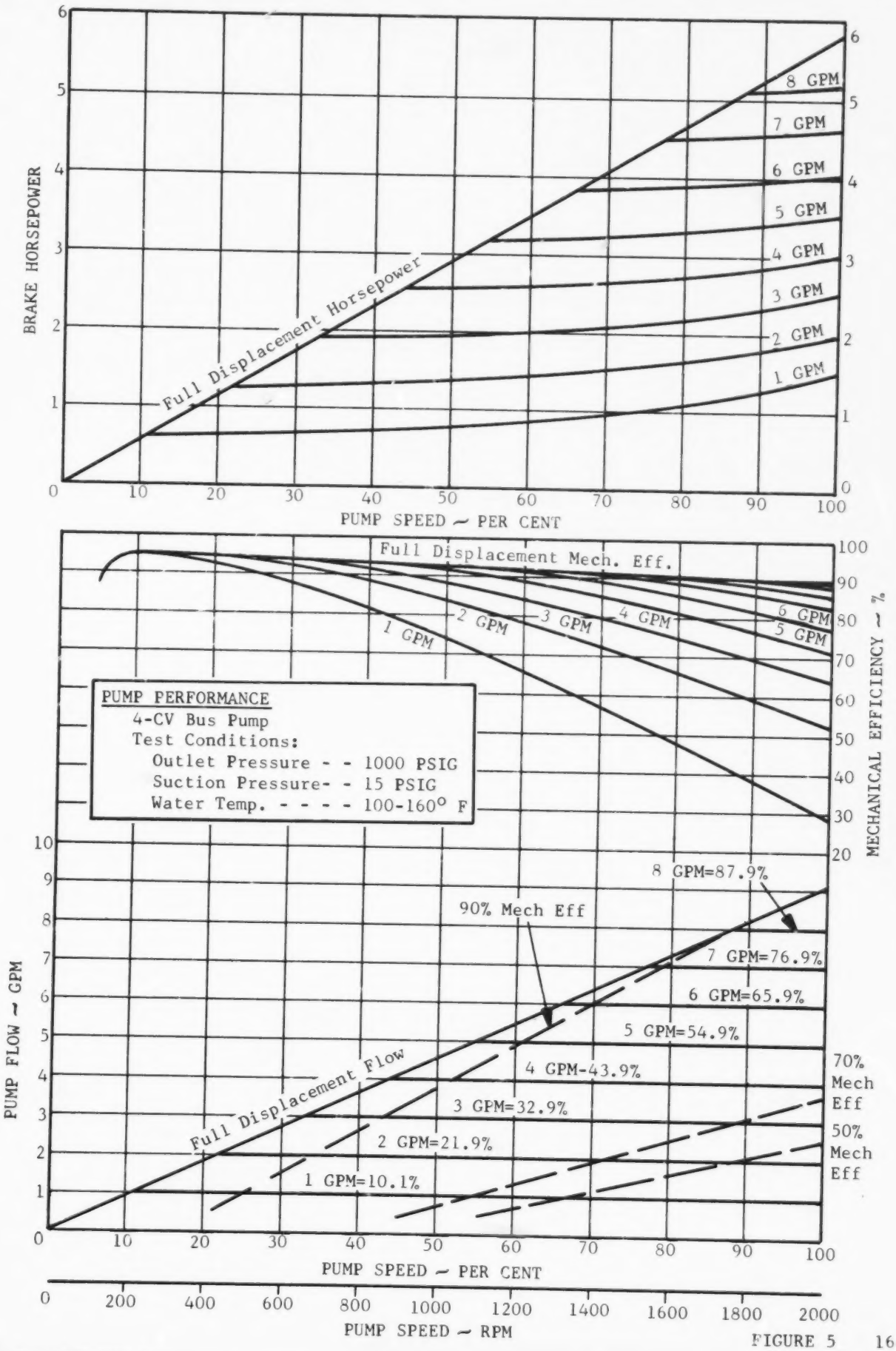

BRAKE HORSEPOWER
Full Displacement Horsepower
8 GPM
7 GPM
6 GPM
5 GPM
4 GPM
3 GPM
2 GPM
1 GPM
PUMP SPEED ~ PER CENT
Full Displacement Mech. Eff.
MECHANICAL EFFICIENCY ~ %
PUMP PERFORMANCE
4-CV Bus Pump
Test Conditions:
Outlet Pressure - - 1000 PSIG
Suction Pressure- - 15 PSIG
Water Temp. - - - - 100-160° F
8 GPM=87.9%
90% Mech Eff
7 GPM=76.9%
6 GPM=65.9%
5 GPM=54.9%
4 GPM-43.9%
3 GPM=32.9%
2 GPM=21.9%
1 GPM=10.1%
70% Mech Eff
50% Mech Eff
PUMP FLOW ~ GPM
Full Displacement Flow
PUMP SPEED ~ PER CENT
PUMP SPEED ~ RPM

FIGURE 5

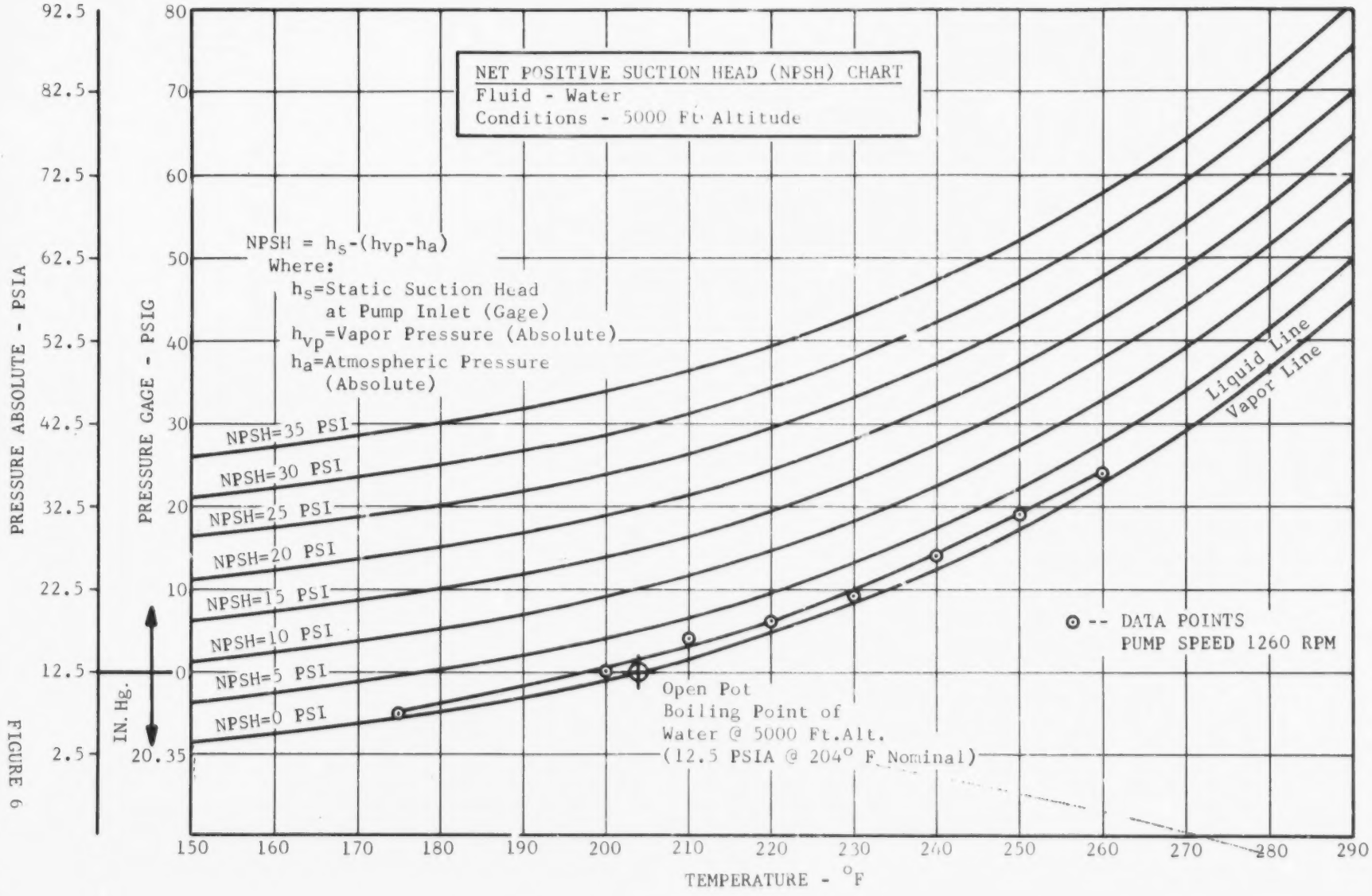

NET POSITIVE SUCTION HEAD (NPSH) CHART
Fluid - Water
Conditions - 5000 Ft. Altitude
NPSH = h_s-(h_{vp}-h_a)
Where:
h_s=Static Suction Head at Pump Inlet (Gage)
h_{vp}=Vapor Pressure (Absolute)
h_a=Atmospheric Pressure (Absolute)
NPSH=35 PSI
NPSH=30 PSI
NPSH=25 PSI
NPSH=20 PSI
NPSH=15 PSI
NPSH=10 PSI
NPSH=5 PSI
NPSH=0 PSI
Liquid Line
Vapor Line
⊙ -- DATA POINTS
PUMP SPEED 1260 RPM
Open Pot
Boiling Point of
Water @ 5000 Ft.Alt.
(12.5 PSIA @ 204° F Nominal)
PRESSURE ABSOLUTE - PSIA
PRESSURE GAGE - PSIG
IN. Hg.
20.35
TEMPERATURE - °F
92.5
82.5
72.5
62.5
52.5
42.5
32.5
22.5
12.5
2.5
80
70
60
50
40
30
20
10
0
150
160
170
180
190
200
210
220
230
240
250
260
270
280
290

FIGURE 6

contract requirements, are also indicated.

Minimum N.P.S.H. characteristics shown were taken directly from Lear Motors feedpump data, corrected only for the individual contractors working fluid specific gravity. It has been our experience, as pointed out in the feedpump design studies, that while the minimum N.P.S.H. characteristics of the Lear Motors feedpump design is low, significant gains in Rankine cycle efficiency are obtained by using an externally driven boost pump designed to handle even lower N.P.S.H. conditions.

D. MATERIAL COMPATIBILITY STUDY

The proposed feedpumps utilize a flexible diaphragm to provide positive hermetic sealing of the working fluid from the atmosphere and oil contamination. Lear Motors Corporation has found that many of the common elastomers demonstrate all of the physical qualities necessary to effect this seal. However, compatibility with the working fluid must be established.

At the first contract coordination meeting, due to the small number of elastomers which had been tested, it was decided that each of the organic fluid system contractors should perform compatibility tests on candidate elastomers and non-metallic materials. Samples of the candidate materials and a suggested test procedure were provided by Lear Motors Corporation.

The compiled results of these tests as well as additional data from the system contractors and an organic fluid manufacturer are shown. A summary of the elastomers which are acceptable for the feedpump sealing diaphragms follows:

THERMO ELECTRON CORPORATION - Organic Working Fluid-Fluorinol 85

Elastomers which have passed initial compatibility tests with the TECO organic working fluid at 250°F.

1. Silicone Rubber, Parker-Compound No. S455-7.

AEROJET LIQUID ROCKET COMPANY - Organic Working Fluid AEF-78

Elastomers which have passed initial compatibility tests with the ALRC organic working fluid at 250°F.

1. Silicon-Rubber - (Dow Corning L-53).
2. Silicon Rubber - (Dow Corning L-63).

3. Chloroprene Rubber - (Dupont "Neoprene").
4. Polysulfide Rubber - (Thiokol Chemical Corp.).

The results of these preliminary tests have demonstrated that elastomers for use with the organic working fluids are available. However, due to the short term and limited scope of these tests, Lear Motors recommends additional investigations.

Results of Material Compatibility Tests Performed by Thermo Electron Corp. [1]

Test Conditions:

Solution - Florinol 85 and oil (10 to 1 ratio)
Temperature - 250°F

LMC Candidate Material	Percent Weight Change	Time (days)	Visual Observations
Chloroprene Rubber	+ 6.8	17	Surface corroded, small pin holes, but still strong. Liquid yellow with small black particles hanging, white particles sticking on glass.
Nitrel	+ 8.3	17	Grey color at bottom in liquid F-85 immersed portion, also softer at bottom. White spots scattered on surface, white line at interface liquid yellow black colored, with some fine white particles hanging.
Polyimide (Dupont) Vespel SP-1	+ .48	6	Color changed to a darker brown, material stron, F-85 and oil light brown, very few suspended fine fibers.
Polyamide DFS-20 (experimental-Dupont)	+ . 11	6	Very few brown spots at bottom, more yellow colored material, slightly softer, oil and F-85 light brown.

CONCLUSIONS

1. Neither Neoprene nor Nitrel Rubber should be used in the presence of F-85 liquid and oil at 250°F.

2. SP-1 Polyimide may be used in the system at 250°F but preferably a lower temperature in the presence of F-85 liquid and oil. It showed about 0.44% swelling.

3. High temperature Polyamide DFS-20 may be used in the system at 250°F or a lower temperature in the presence of F-85 liquid and oil. It showed about 1.1% swelling with very light softening.

[1] Based on personal correspondence between Luco R. DiNanno, Development Engineer for Thermo Electron, Corp, and writer, September 13, 1972.

Results of Material Compatibility Tests Performed by Thermo Electron Corp. [2]

Test Conditions:

Solution -	Fluorinol 85 and lubricating oil 10 to 1 ratio
Temperature -	250°F
Duration -	17 days

Material	Visual Observation
Dow Corning, LS53, Fluorosilicone Rubber (-2" x 0.07" O.D.)	The rubber was much softer, lightly attack at bottom in liquid immersed portion, brownish material was sticking on glass and liquid F-85 turned to very light brown.
Dow Corning LS63, Fluorosilicone Rubber (1.5" x 0.1" O.D.)	The rubber was much softer, lightly attack in liquid immersed portion. F-85 turned very light brown.
Parker No. S455-7 Silicon Rubber (2" x 0.1" O.D.)	The rubber was lightly softer, no apparent change, F-85 turned light brown.

RECOMMENDATIONS:

Silicone Rubber - Parker No. S455-7 is recommended to be used in the system at temperature not exceeding 250°F in the presence of F-85 solution and air.

[2] Based on correspondence between Dyer Kenney, Contract Project Officer for Environmental Protection Agency, and writer October, 1972.

Material Compatibility Information on Elastomers Obtained from the Fluid Manufacturer - Halocarbon Products Corporation [3]

Test Conditions:

Solution - Fluorinol 85
Temperature - 284°F
Duration 7 days

Material	% Weight Change	% Thickness Change
Ethylene-Propylene Rubber	1.3 to 2.6	1.1 to 1.5
Silicone Rubber	4.7 to 11	2.6 to 3.4
Neoprene	15 to 16	5.2 to 10
Pure Gum & Polyisoprene	20 to 46	8.8 to 16
Flurosilicone Rubber	1.6 to 49	-23 to 18
Viton A	34 to 54	13 to 15
Butyl	16	7.0

Completely Unsatisfactory: Buna N, Buna S and Hypalon.

[3]Based on correspondence between Mr. Rex Conner, Halocarbon Products Corporation and writer, August, 1972.

Results of Material Compatibility Tests Performed by Aerojet Liquid Rocket Co. [4]

Test Conditions:

Solution - AEF-78 (ALRC working fluid)
Temperature - 250°F
Duration - 150 hours

LMC Candidate Materials	Test Results
Neoprene (Diaphragm Stock)	Marginal
Delrin (Dupont)	Not Acceptable
Vespel SP-1 Polyimide (Dupont)	Good
Vespel SP-21 Polyimide (Dupont)	Good
Polyamide DFS-21, Experimental (Dupont)	Good
Nitrel (Diaphragm Stock)	Not Acceptable
Viton (Diaphragm Stock)	Marginal
Polyamide DFS-20, experimental (Dupont)	Good
Natural Rubber (A.B. Boyd)	Not Acceptable
Silicone	Good
Polyurethane (Newage Industries)	Marginal

[4]Based on telephone conversation between A.H. Kreeger, Manager Automotive Rankine Program, Aerojet Liquid Rocket Co. and writer, November, 1972.

Results of Elastomer Material Compatibility Tests Performed by Aerojet Liquid Rocket Co. [5]

Test Conditions:

Solution -	AEF-78
Temperature -	250°F
Duration	Long Term

Elastomer	Test Results
Silicon Rubber (Dow Corning L-53)	Good
Silicon Rubber (Dow Corning L-63)	Good
Silicon Rubber (Parker HS 455-7)	Good
Florosilicon Rubber	Not Acceptable
Chloroprene Rubber	Good
Polysulfide (Thiokol Chemical Corp.)	Good

[5]Based on telephone conversation between A.H. Kreeger, Manager Automotive Rankine Program, Aerojet Liquid Rocket Co. and writer, September, 1972.

CANDIDATE MATERIALS FOR SYSTEM

CONTRACTOR LMC PROPOSED FEEDPUMPS

Materials for Working Fluid Section of Feedpump

Component	SES	ALRC	TECO
Manifold (Inlet & Outlet)	Cast 356 Aluminum	Cast 356 Aluminum	Cast 356 Aluminum
Valve Plate	Stainless Steel Type 303	Stainless Steel Type 303	Stainless Steel Type 303
Outlet Valve	Aluminum 6061-T6	Aluminum 6061-T6	Aluminum 6061-T6
Inlet Valve	Aluminum 6061-T6	Aluminum 6061-T6	Aluminum 6061-T6
Valve Spring (Inlet & Outlet)	Stainless Steel Type 804	Stainless Steel Type 804	Stainless Steel Type 804
Seals (Enclosed & Static)	Nitrile Rubber	Silicone Rubber	Silicone Rubber
Diaphragm Seal	"Hydrin" Rubber	1. Silicone Rubber* 2. Polysulfide Rubber*	Silicone Rubber*

Material for Oil Lubricate Section of Feedpump

Feedpump Housing	1. Cast Ductile Iron 2. Cast Aluminum 356-T6**	1. Cast Ductile Iron 2. Cast Aluminum 356-T6**	1. Cast Ductile Iron 2. Cast Aluminum 356-T6**
Bearings (Drive Shaft)	Steel (Anti-Friction)	Steel (Anti-Friction)	Steel (Anti-Friction)
Drive Shaft	Carborized 4620 Steel	Carborized 4620 Steel	Carborized 4620 Steel
Pistons	Carborized 1117 Steel	Carborized 1117 Steel	Carborized 1117 Steel
Bearings (Piston)	Aluminum Bronze	Aluminum Bronze	Aluminum Bronze
Cylinders	Cast Nodular Iron	Cast Nodular Iron	Cast Nodular Iron
Seals	Buna "N" Rubber	Buna "N" Rubber	Buna "N" Rubber

* Preliminary selection based on limited material compatibility study.

** Additional stress analysis required.

TABLE 3

STEAM ENGINE SYSTEMS TECHNICAL REQUIREMENTS AND LEAR MOTORS CORP. APPROACH

VAPOR GENERATOR FEEDPUMP FOR RANKINE CYCLE AUTOMOTIVE PROPULSION SYSTEM

EPA CONTRACT NUMBER 68-01-0437

December 1972

STEAM ENGINE SYSTEMS

PUMP PERFORMANCE

TECHNICAL REQUIREMENTS

1. Expander Speed The speed of the reciprocating expander during operation will vary from a minimum of 300 rpm to a maximum of 2500 rpm.

2. Flow The flow range shall be from 0 to 2.5 gpm. Maximum system flow (2.5 GPM) shall occur at 40 percent of maximum expander speed. The feedpump shall be capable of modulating upon command from the propulsion control system to satisfy any flow condition within the the operating range specified. The feedpump shall be capable of operating with no inlet flow for periods up to one minute without damage.

3. Discharge Pressure The feedpump shall be capable of delivering flow as specified, while discharging to a pressure of 1200 psia. The discharge pressure shall be stable with no high frequency oscillations in excess of 25 psia.

4. Efficiency The feedpump mechanical efficiency shall be 70% or greater over the flow range from 30% to 80% of maximum design flow. The mechanical efficiency shall be 50% or greater at 10% of maximum design flow.

5. Fluid The working fluid for this system will be demineralized water.

TECHNICAL APPROACH

The estimated performance curves, on the following page, demonstrate how the proposed LMC feedpump would meet the SES technical requirements shown above. This estimate was obtained by extrapolating actual LMC feedpump data.

The maximum pressure ripple requirement is met by utilizing an odd number of cylinders which are sized so that the instantaneous total flow of the pump for any given crank angle is as smooth as possible.

Additional information regarding the LMC technical approach to the SES feedpump performance requirements may be found in the technical discussions section of this report.

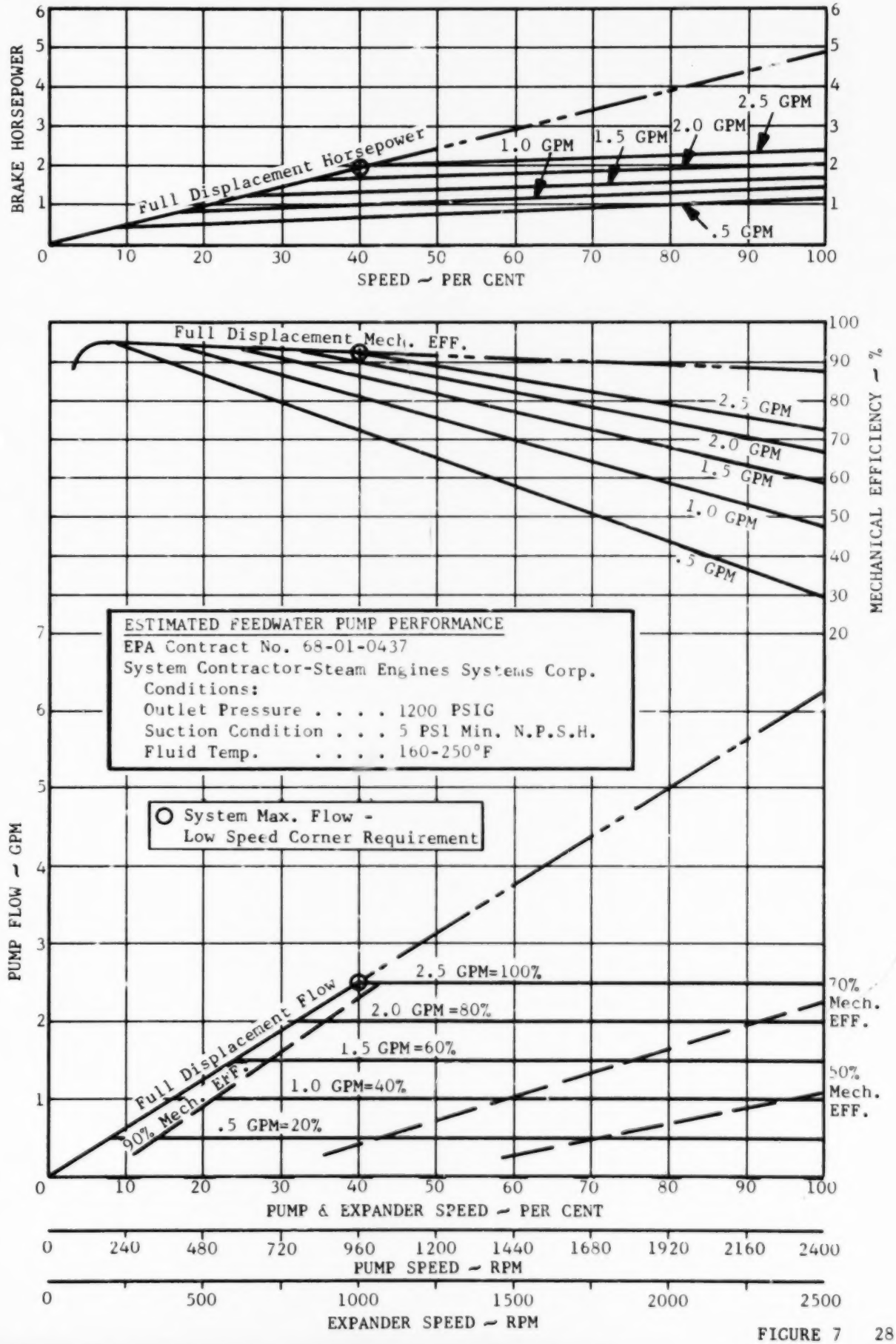

BRAKE HORSEPOWER
Full Displacement Horsepower
1.0 GPM
1.5 GPM
2.0 GPM
2.5 GPM
.5 GPM
SPEED ~ PER CENT
Full Displacement Mech. EFF.
2.5 GPM
2.0 GPM
1.5 GPM
1.0 GPM
.5 GPM
MECHANICAL EFFICIENCY ~ %
ESTIMATED FEEDWATER PUMP PERFORMANCE
EPA Contract No. 68-01-0437
System Contractor-Steam Engines Systems Corp.
Conditions:
Outlet Pressure . . . . 1200 PSIG
Suction Condition . . . 5 PSI Min. N.P.S.H.
Fluid Temp. . . . . 160-250°F
System Max. Flow -
Low Speed Corner Requirement
PUMP FLOW ~ GPM
Full Displacement Flow
90% Mech. Eff.
2.5 GPM=100%
2.0 GPM=80%
1.5 GPM=60%
1.0 GPM=40%
.5 GPM=20%
70% Mech. EFF.
50% Mech. EFF.
PUMP & EXPANDER SPEED ~ PER CENT
PUMP SPEED ~ RPM
EXPANDER SPEED ~ RPM

FIGURE 7

STEAM ENGINE SYSTEMS

PUMP INLET ENVELOPE

TECHNICAL REQUIREMENTS

1. Inlet Pressure The minimum pressure at the pump inlet valve during start-up maybe as low as 0.09 psia. During system operation, the inlet pressure may vary from a minimum of 7 psia to a maximum of 50 psia.

2. Net Positive Suction Head The net positive suction head required by the feedpump shall be minimized so that the pump can operate without cavitating under low liquid head conditions without the necessity of a boost pump.

3. Temperature Range During system operation, the temperature of the working fluid may vary from 180°F to 250°F. At system start-up the temperature of the working fluid may vary from 32°F to 275°F.

4. Temperature Transients The feedpump shall stand feedwater temperature transients of 3°F per second for periods of 10 seconds without cavitating.

5. Booster Pump A booster pump for providing the required net positive suction head to the feedpump shall be considered only if trade-off studies conducted by the contractor reveal that the utilization of a boost pump will simplify the feedpump design sufficiently to offset the additional cost, weight and power consumption required by the boost pump.

TECHNICAL APPROACH

The estimated pump inlet envelope on the following page shows the Lear pump capability while operating at the 85% to 90% volumetric efficiency level. The N.P.S.H. values shown are taken from actual data.

RECOMMENDATIONS

Due to the inherent lower N.P.S.H. characteristics of a centrifugal pump and its ability to handle large quantities of entrained vapor, a separate centrifugal boost pump is recommended for this system. The size and characteristics of this pump will require a trade-off study of the system condenser capability and the amount of sub-cooling allowable before significant decreases in cycle efficiency occur.

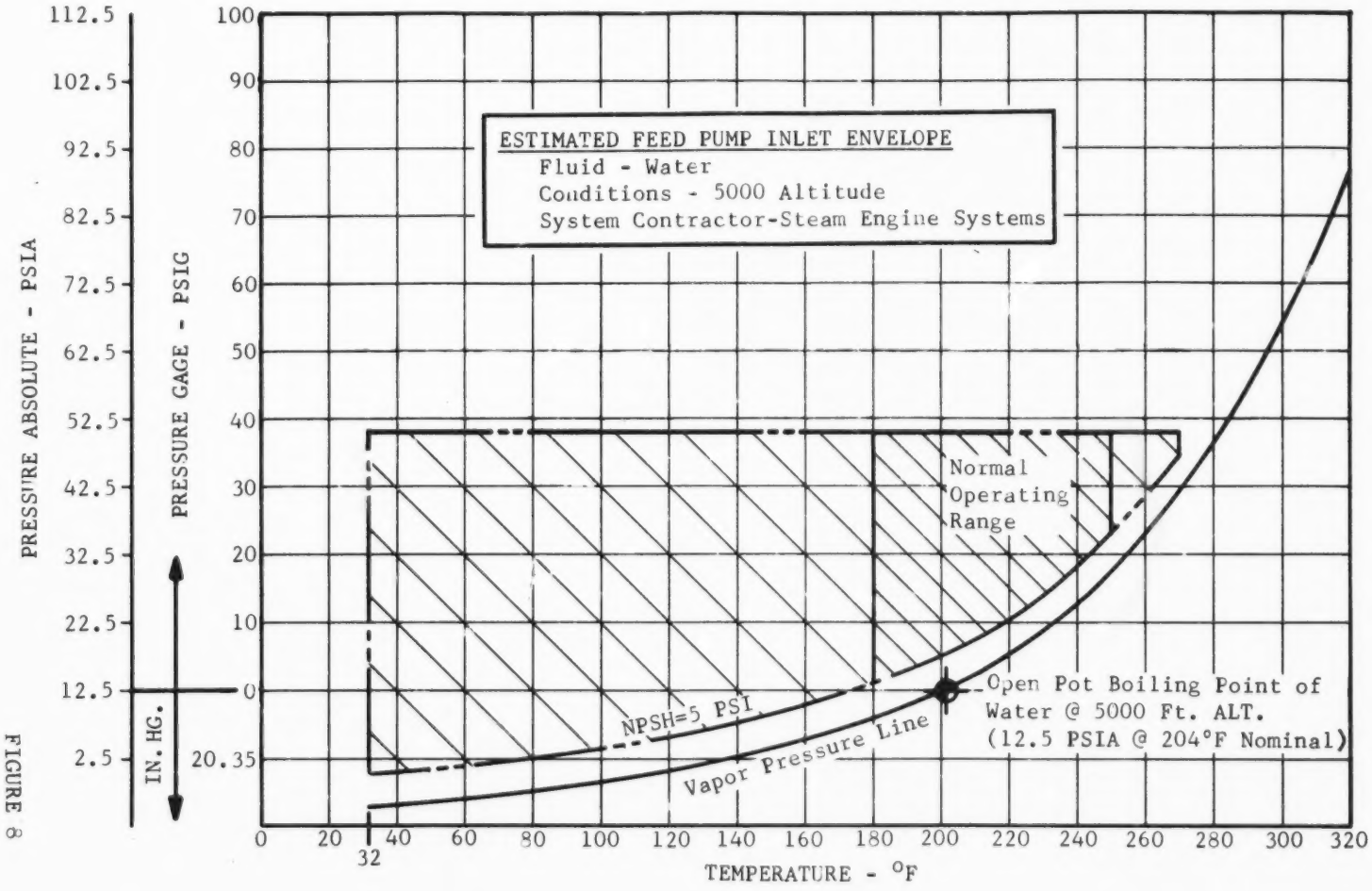

ESTIMATED FEED PUMP INLET ENVELOPE
Fluid - Water
Conditions - 5000 Altitude
System Contractor-Steam Engine Systems
Normal Operating Range
NPSH=5 PSI
Vapor Pressure Line
Open Pot Boiling Point of Water @ 5000 Ft. ALT. (12.5 PSIA @ 204°F Nominal)
PRESSURE ABSOLUTE - PSIA
PRESSURE GAGE - PSIG
IN. HG.
20.35
112.5
102.5
92.5
82.5
72.5
62.5
52.5
42.5
32.5
22.5
12.5
2.5
100
90
80
70
60
50
40
30
20
10
0
0 20 32 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320
TEMPERATURE - °F

FIGURE 8

STEAM ENGINE SYSTEMS

PUMP DRIVE

TECHNICAL REQUIREMENTS

1. Drive Mechanism The feedpump shall be mechanically driven with power taken from the expander shaft or gearbox. The feedpump shall provide the total range of required flow and pressure with the expander operation at any speed within its specified speed range. The non-linear relationship between expander speed and required feedpump flow and pressure shall be accommodated by variable drive mechanism characteristics and/or variable feedpump and drive mechanism should be capable of being mounted in the power plant as required by the system contractor.

TECHNICAL APPROACH

An outline drawing of the proposed Lear Motors Pump is shown on the following page. The pump has variable displacement capability and can be directly driven by the expander. Pump to expander speed ratio and interface are not firm and will have to be worked out between Lear Motors and the system contractor.

Pump physical size is based on the contracts pump performance requirements, i.e. flow rate, and any change in the design flow rate will directly affect pumps physical size and weight.

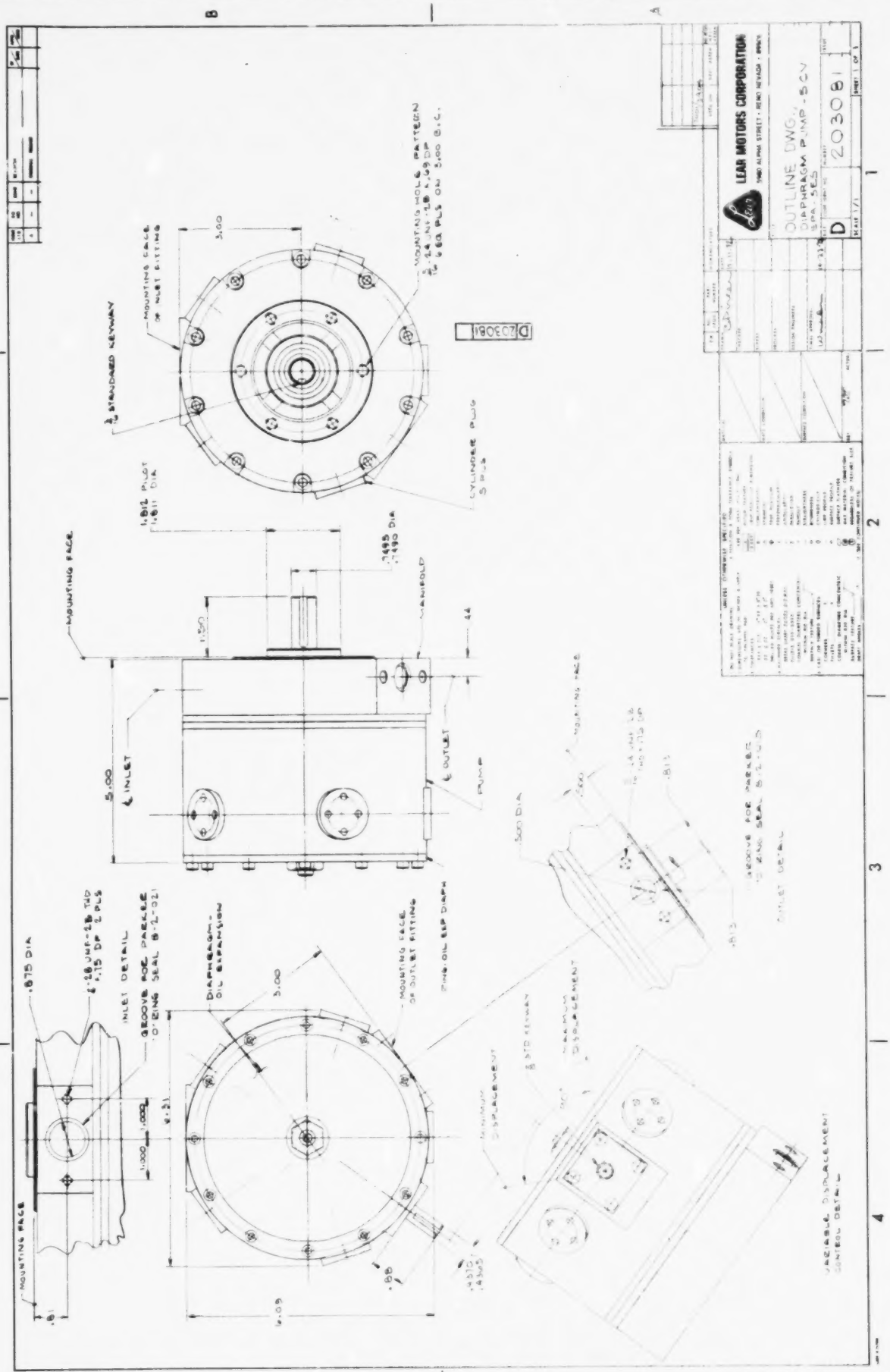

LEAR MOTORS CORPORATION
OUTLINE DWG.
DIAPHRAGM PUMP - SCV
SPA-565
D 203081
MOUNTING FACE
INLET
OUTLET
PUMP
MANIFOLD
MOUNTING HOLES PATTERN
STANDARD KEYWAY
CYLINDER PLUG
INLET DETAIL
GROOVE FOR PARKER O-RING SEAL 8-2-021
DIAPHRAGM-OIL EXPANSION
MOUNTING FACE OF OUTLET FITTING
RING OIL EXP DIAPH
MAXIMUM DISPLACEMENT
MINIMUM DISPLACEMENT
VARIABLE DISPLACEMENT CONTROL DETAIL

FIGURE 9

STEAM ENGINE SYSTEMS

GENERAL REQUIREMENTS

TECHNICAL REQUIREMENTS

1. Vehicle Design Goals Where applicable, "Vehicle Design Goals-Six Passenger Automobile", shall be used as a guide for the design requirements.

2. Materials All materials utilized in the construction of the feedpump shall be corrosive resistant in a water/air environment.

3. Lubrication Any lubricants that may be used within the feedpump and its drive mechanism shall not mix with or contaminate the working fluid. The lubricant shall be the same as that used in the expander.

4. Leakage There shall be no external leakage from the feedpump and its drive mechanism. The unit shall be capable of being hermetically sealed.

5. Working Fluid Drainage The feedpump shall be designed such that all working fluid within the pump will drain back out of the inlet side of the pump into the system sump at shutdown. Care should be taken to assure that all pockets of fluid are adequately drained to prevent damage due to freezing.

TECHNICAL APPROACH

The Lear Motors pump design, with its flexible diaphragm seal provides inherent hermetic sealing of working fluid from the air and lubricated section of the pump.

Working fluid drainage requirement will be met by unseating the inlet valves at system shutdown allowing the fluid in each chamber to drain back to the inlet manifold. This approach would not degrade the pumps performance and would be integral to the pump.

Materials used for the construction of the proposed SES feedpump are shown in Table 3. The material used for the pumps flexible sealing member can be constructed from almost any elastometer which can meet the temperature requirements and is compatible with the working fluid.

AEROJET LIQUID ROCKET COMPANY TECHNICAL REQUIREMENTS AND LEAR MOTORS CORP. APPROACH

VAPOR GENERATOR FEEDPUMP FOR RANKINE CYCLE AUTOMOTIVE PROPULSION SYSTEM

EPA CONTRACT NUMBER 68-01-0437

DECEMBER 1972

AEROJET LIQUID ROCKET COMPANY

PUMP PERFORMANCE

TECHNICAL REQUIREMENTS

1. Speed Range The speed of the turbine expander will vary from a minimum of 16,800 rpm to a maximum of 31,200 rpm. This same speed ratio (1.85:1) will be available from the expander gearbox, if required.

2. Flow The feedpump and its drive mechanism shall be capable of delivering a constant flow of organic fluid from a minimum of 0.5 gpm to a maximum of 28 gpm at a discharge pressure from 650 psia to 1100 psia and at any given expander speed within the range specified. Maximum system flow (28 gpm) shall occur at 59 percent maximum expander speed.

3. Discharge Pressure The feedpump shall be capable of delivering flow as specified while discharging to a pressure of 1100 psia. The discharge pressure shall be stable with no high frequency oscillations in excess of 25 psi.

4. Efficiency The feedpump mechanical efficiency shall be 70% or greater over the flow range from 30% of maximum design flow to 80% of maximum design flow. The mechanical efficiency shall be 50% or greater at 10% of maximum design flow.

5. Fluid The working fluid will be a non-corrosive organic fluid with a density of approximately 100 lbs/ft^3.

TECHNICAL APPROACH

The estimated performance curves, on the following page, demonstrate how the proposed LMC feedpump would meet the ALRC technical requirements shown above. This estimate was obtained by extrapolating actual LMC feedpump data.

The maximum pressure ripple requirement is met by utilizing an odd number of cylinders which are sized so that the instantaneous total flow of the pump for any given crank angle is as smooth as possible.

Additional information regarding the LMC technical approach to the ALRC feedpump performance requirements may be found in the technical discussions section of this report.

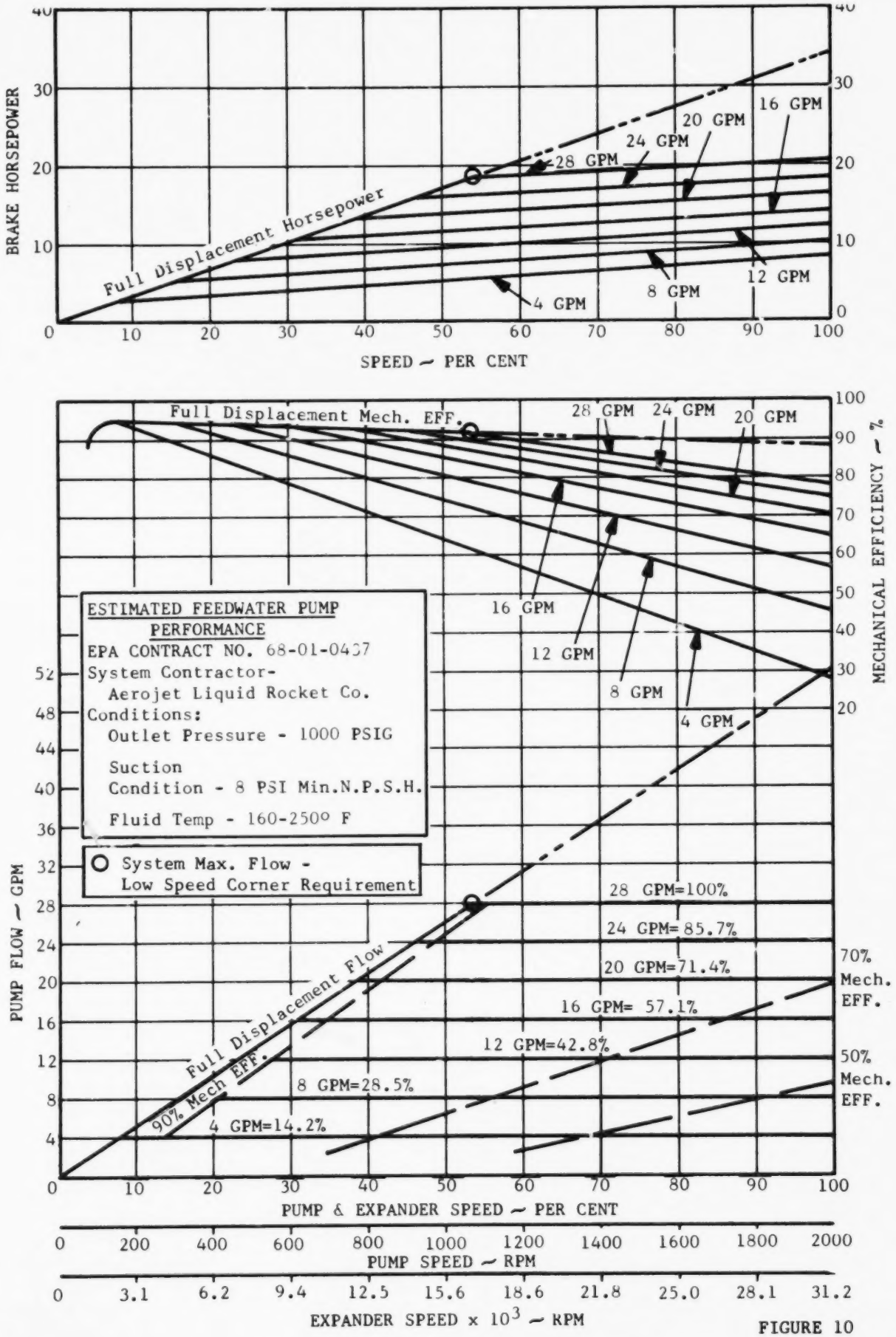

BRAKE HORSEPOWER
Full Displacement Horsepower
28 GPM
24 GPM
20 GPM
16 GPM
12 GPM
8 GPM
4 GPM
SPEED ~ PER CENT
Full Displacement Mech. EFF.
28 GPM 24 GPM 20 GPM
16 GPM
12 GPM
8 GPM
4 GPM
MECHANICAL EFFICIENCY ~ %
ESTIMATED FEEDWATER PUMP PERFORMANCE
EPA CONTRACT NO. 68-01-0437
System Contractor-
Aerojet Liquid Rocket Co.
Conditions:
Outlet Pressure - 1000 PSIG
Suction
Condition - 8 PSI Min.N.P.S.H.
Fluid Temp - 160-250° F
System Max. Flow -
Low Speed Corner Requirement
28 GPM=100%
24 GPM= 85.7%
20 GPM=71.4%
16 GPM= 57.1%
12 GPM=42.8%
8 GPM=28.5%
4 GPM=14.2%
Full Displacement Flow
90% Mech EFF.
70% Mech. EFF.
50% Mech. EFF.
PUMP FLOW ~ GPM
PUMP & EXPANDER SPEED ~ PER CENT
PUMP SPEED ~ RPM
EXPANDER SPEED x 10^3 ~ RPM

FIGURE 10

AEROJET LIQUID ROCKET COMPANY

PUMP INLET ENVELOPE

TECHNICAL REQUIREMENTS

1. Inlet Pressure The minimum pressure at the pump inlet valve during start-up may be as low as 1.0 psia. During system operation, the inlet pressure may vary from a minimum of 10 psia to a maximum of 40 psia.

2. Net Positive Suction Head The minimum liquid head at the pump inlet is 7 inches. Care must be taken to assure cavitation does not occur during start-up or during abnormal operating conditions resulting in a loss of condenser sub-cooling.

3. Temperature Range During system operation, the temperature of the working fluid may vary from 160°F to 250°F. At start-up the temperature of the working fluid may vary from -40°F to 275°F.

4. Booster Pump A booster pump for providing the required net positive suction head to the feedpump shall be considered only if trade-off studies conducted by the contractor reveal that the utilization of a boost pump will simplify the feedpump design sufficiently to offset the additional cost, weight,and power consumption required by the boost pump.

TECHNICAL APPROACH

The estimated pump inlet envelope on the following page shows the Lear pump capability while operating at the 85% to 90% volumetric efficiency level. The N.P.S.H. values shown are taken from actual data which has been corrected to meet the system contractors fluid characteristics.

RECOMMENDATIONS

Due to the inherent lower N.P.S.H. characteristics of a centrifugal pump and its ability to handle large quantities of entrained vapor, a separate centrifugal boost pump is recommended for this system. The size and characteristics of this pump will require a trade-off study of the systems condenser capability and the amount of sub-cooling allowable before significant decreases in cycle efficiency occur.

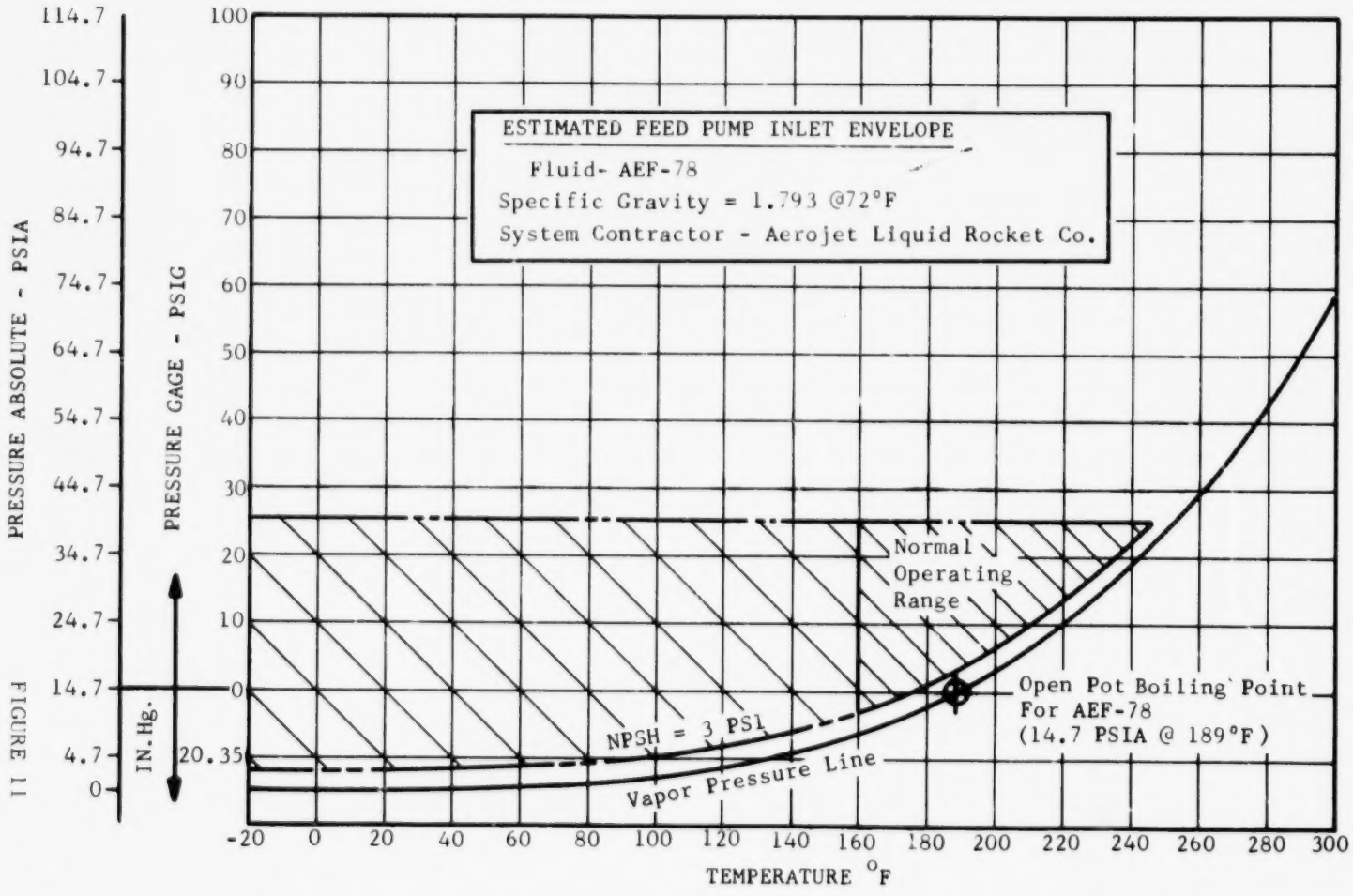

ESTIMATED FEED PUMP INLET ENVELOPE
Fluid- AEF-78
Specific Gravity = 1.793 @72°F
System Contractor - Aerojet Liquid Rocket Co.
PRESSURE ABSOLUTE - PSIA
PRESSURE GAGE - PSIG
IN. Hg.
20.35
Normal Operating Range
NPSH = 3 PSI
Vapor Pressure Line
Open Pot Boiling Point For AEF-78 (14.7 PSIA @ 189°F)
TEMPERATURE °F

FIGURE 11

AEROJET LIQUID ROCKET COMPANY

PUMP DRIVE AND CONTROL

TECHNICAL REQUIREMENTS

1. Drive Mechanism The feedpump shall be mechanically driven with power taken from the expander shaft or gearbox. The feedpump shall provide the total range of required flow and pressure with the expander operation at any speed within its specified speed range. The non-linear relationship between expander speed and required feedpump flow and pressure shall be accommodated by variable drive mechanism characteristics and/or variable feedpump capacity characteristics. The feedpump and drive mechanism should be capable of being mounted in the power plant as required by the system contractor.

2. Control Movement of the vehicle accelerator pedal will provide a mechanical input signal to the feedpump for flow modulation.

TECHNICAL APPROACH

An outline drawing of the proposed Lear Motors pump is shown on the following page. The pump has variable displacement capability and can be directly driven by the expander. Pump to expander speed ratio and interface are not firm and will have to be worked out between Lear Motors and the system contractor.

Pump physical size is based on the contract pump performance requirements, i.e. flow rate, and any change in the design flow rate will directly affect pump physical size and weight.

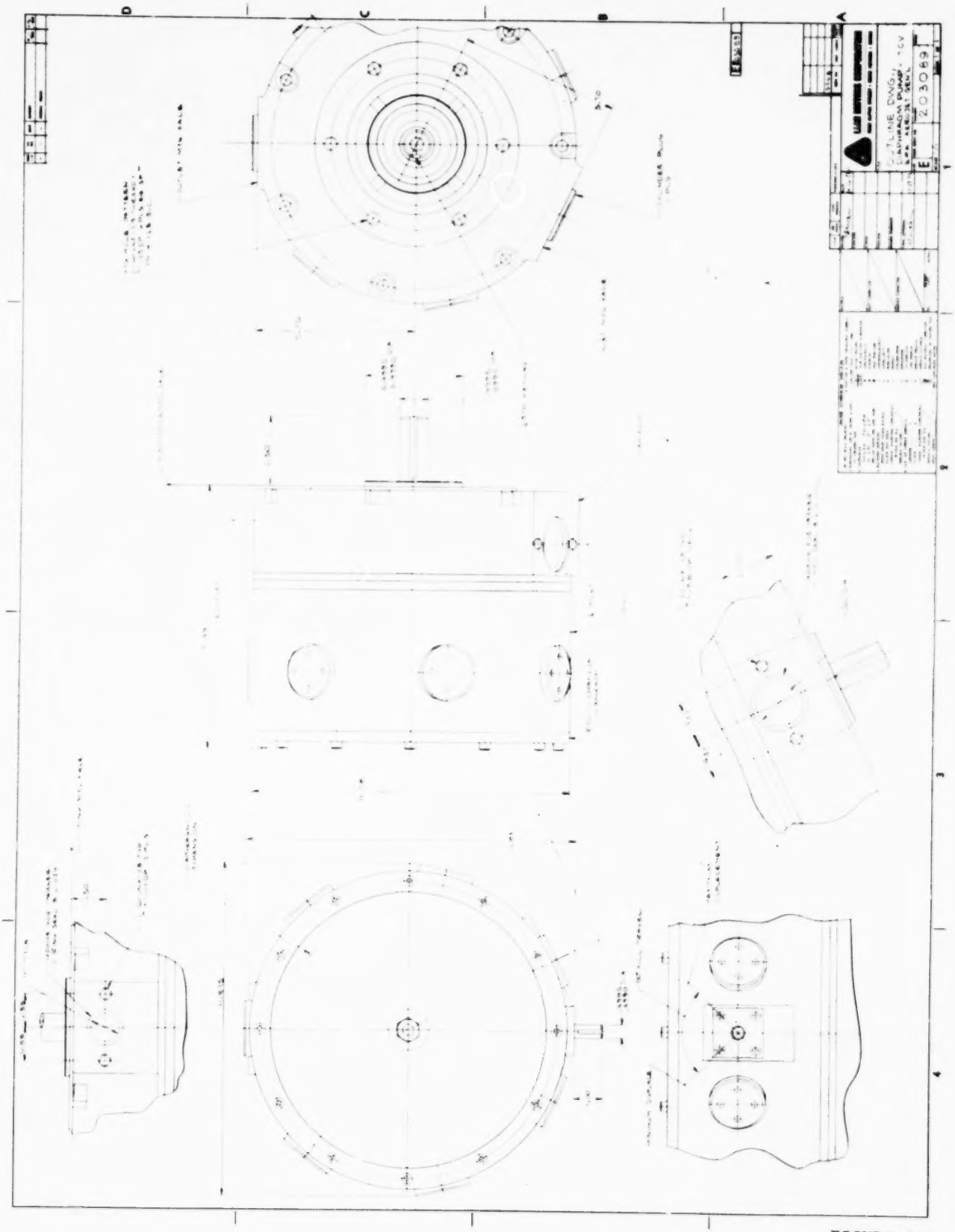


FIGURE 12

AEROJET LIQUID ROCKET COMPANY

GENERAL REQUIREMENTS

TECHNICAL REQUIREMENTS

1. Vehicle Design Goals Where applicable, "Vehicle Design Goals-Six Passenger Automobile", shall be used as a guide for the design requirements.

2. Leakage There shall be no external leakage from the feedpump and its drive mechanism. The unit shall be capable of being hermetically sealed.

3. Lubrication Any lubricants that may be used within the feedpump and its drive mechanism shall not mix or contaminate the working fluid. The lubricant shall be the same as that used in the expander.

4. Materials Compatibility All materials utilized in the construction of the feedpump shall not corrode or otherwise chemically react with the fluid in any manner.

TECHNICAL APPROACH

The Lear Motors pump design with its flexible diaphragm seal provides inherent hermetic sealing of working fluid from the air and lubricated section of the pump.

Materials used for the construction of the proposed ALRC feedpump are shown in Table 3. The material used for the pumps flexible sealing member can be constructed from elastomers which meet the temperature requirements and is compatible with the working fluid.

THERMO ELECTRON CORPORATION TECHNICAL REQUIREMENTS AND LEAR MOTORS CORP. APPROACH

VAPOR GENERATOR FEEDPUMP FOR RANKINE CYCLE AUTOMOTIVE PROPULSION SYSTEM

EPA CONTRACT NUMBER 68-01-0437

DECEMBER 1972

THERMO ELECTRON CORPORATION

PUMP PERFORMANCE

TECHNICAL REQUIREMENTS

1. Expander Speed Range The operating speed range of the reciprocating expander will vary from a minimum of 300 rpm to a maximum of 1800 rpm.

2. Flow Flow range shall be from 0 to 17 gpm. Maximum system flow (17 gpm) shall occur at 45 percent maximum expander speed. The feedpump shall be capable of modulating upon command from the propulsion control system to satisfy any flow condition within the operating range specified.

3. Discharge Pressure The feedpump shall be capable of operating at a constant discharge pressure of 1000 psia while modulating flow over the range specified. The discharge pressure shall be stable with no high frequency oscillations in excess of 25 psi.

4. Efficiency The feedpump mechanical efficiency shall be 70% or greater over the flow range from 30% of maximum design flow to 80% of maximum design flow. The mechanical efficiency shall be 50% or greater at 10% of maximum design flow.

5. Fluid The working fluid will be a non-corrosive organic fluid with a density of approximately 80 lbs/ft^3.

TECHNICAL APPROACH

The estimated performance curves, on the following page, demonstrate how the proposed LMC feedpump would meet the TECO technical requirements shown above. This estimate was obtained by extrapolating actual LMC feedpump data.

The maximum pressure ripple requirement is met by utilizing an odd number of cylinders which are sized so that the instantaneous total flow of the pump for any given crank angle is as smooth as possible.

Additional information regarding the LMC technical approach to the TECO feedpump performance requirements may be found in the technical discussions section of this report.

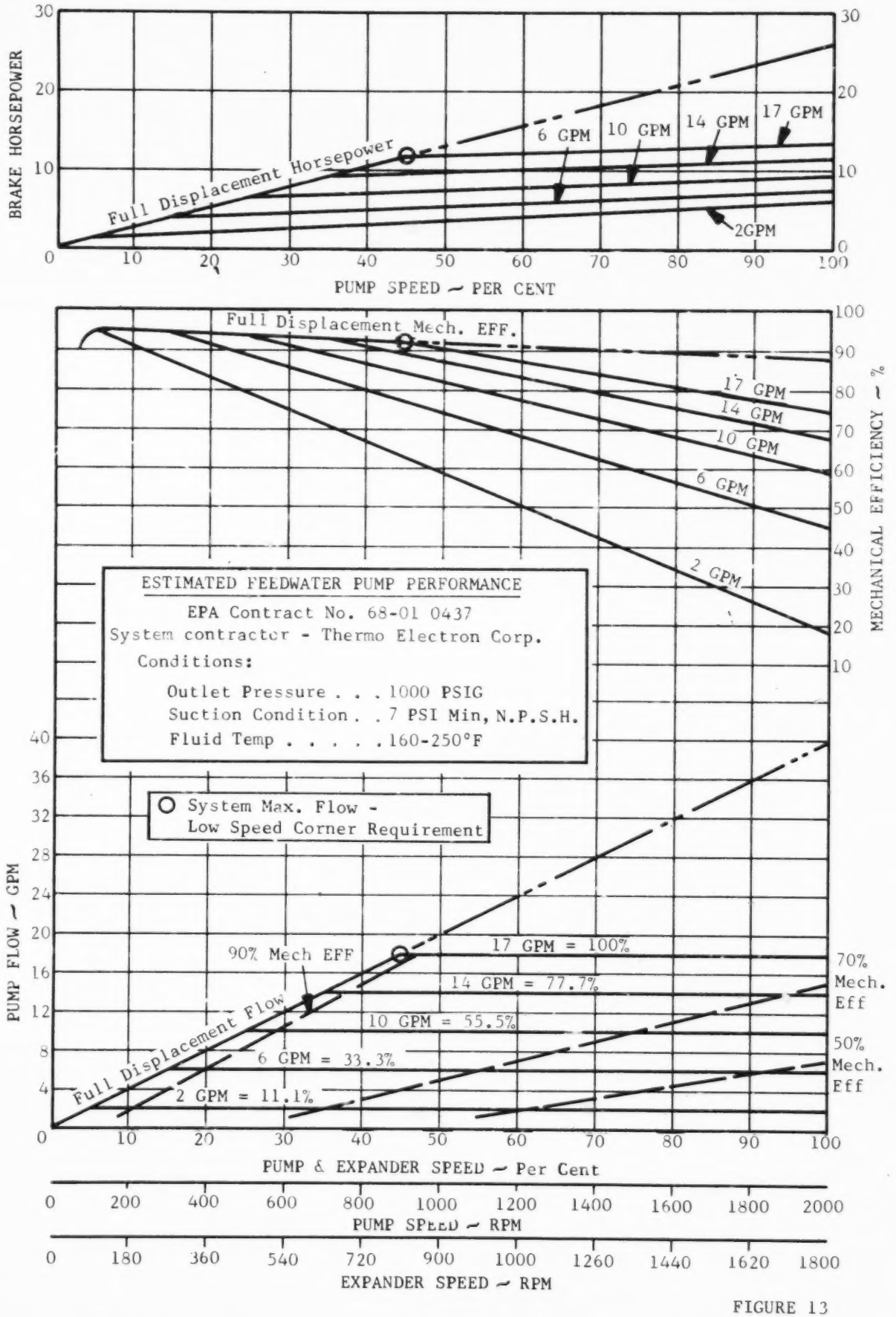

BRAKE HORSEPOWER
Full Displacement Horsepower
6 GPM
10 GPM
14 GPM
17 GPM
2GPM
PUMP SPEED ~ PER CENT
Full Displacement Mech. EFF.
17 GPM
14 GPM
10 GPM
6 GPM
2 GPM
MECHANICAL EFFICIENCY ~ %
ESTIMATED FEEDWATER PUMP PERFORMANCE
EPA Contract No. 68-01 0437
System contractor - Thermo Electron Corp.
Conditions:
Outlet Pressure . . . 1000 PSIG
Suction Condition . . 7 PSI Min, N.P.S.H.
Fluid Temp . . . . . 160-250°F
System Max. Flow -
Low Speed Corner Requirement
PUMP FLOW ~ GPM
90% Mech EFF
17 GPM = 100%
14 GPM = 77.7%
10 GPM = 55.5%
6 GPM = 33.3%
2 GPM = 11.1%
Full Displacement Flow
70% Mech. Eff
50% Mech. Eff
PUMP & EXPANDER SPEED ~ Per Cent
PUMP SPEED ~ RPM
EXPANDER SPEED ~ RPM

FIGURE 13

THERMO ELECTRON CORPORATION

PUMP INLET ENVELOPE

TECHNICAL REQUIREMENTS

1. Inlet Pressure In the Rankine cycle system, the feedpump inlet pressure is determined by the condensing pressure and the elevation of the pump relative to the condenser. At system start-up, prior to the time when condenser pressure builds up due to heat load, the inlet pressure at the pump may be no greater than 0.5 psia. During operation the inlet pressure may vary from a minimum of 5 psia to a maximum of 90 psia.

2. Net Positive Suction Head The minimum liquid head at the pump inlet is 10 inches. Care must be taken to assure cavitation does not occur during start-up or during abnormal operating conditions resulting in a loss of condenser sub-cooling.

3. Temperature Range During system operation, the temperature of the working fluid may vary from 160°F to 250°F. At start-up the temperature of the working fluid may vary from -40°F to 275°F.

4. Booster Pump A booster pump for providing the required net positive suction head to the feedpump shall be considered only if utilization of a boost pump will simplify the feedpump design sufficiently to offset the additional cost, weight and power consumption required by the boost pump.

TECHNICAL APPROACH

The estimated pump inlet envelope on the following page shows the proposed LMC pump capability while operating at the 85% to 90% volumetric efficiency level. The N.P.S.H. values shown are taken from actual data which has been corrected to meet the TECO organic fluid characteristics.

Due to the inherent lower N.P.S.H. characteristics of a centrifugal pump and its ability to handle large quantities of entrained vapor, a separate centrifugal boost pump is recommended for this system. The size and characteristics of this pump will require a trade-off study of the system condenser capability and the amount of sub-cooling allowable before significant decreases in cycle efficiency occur.

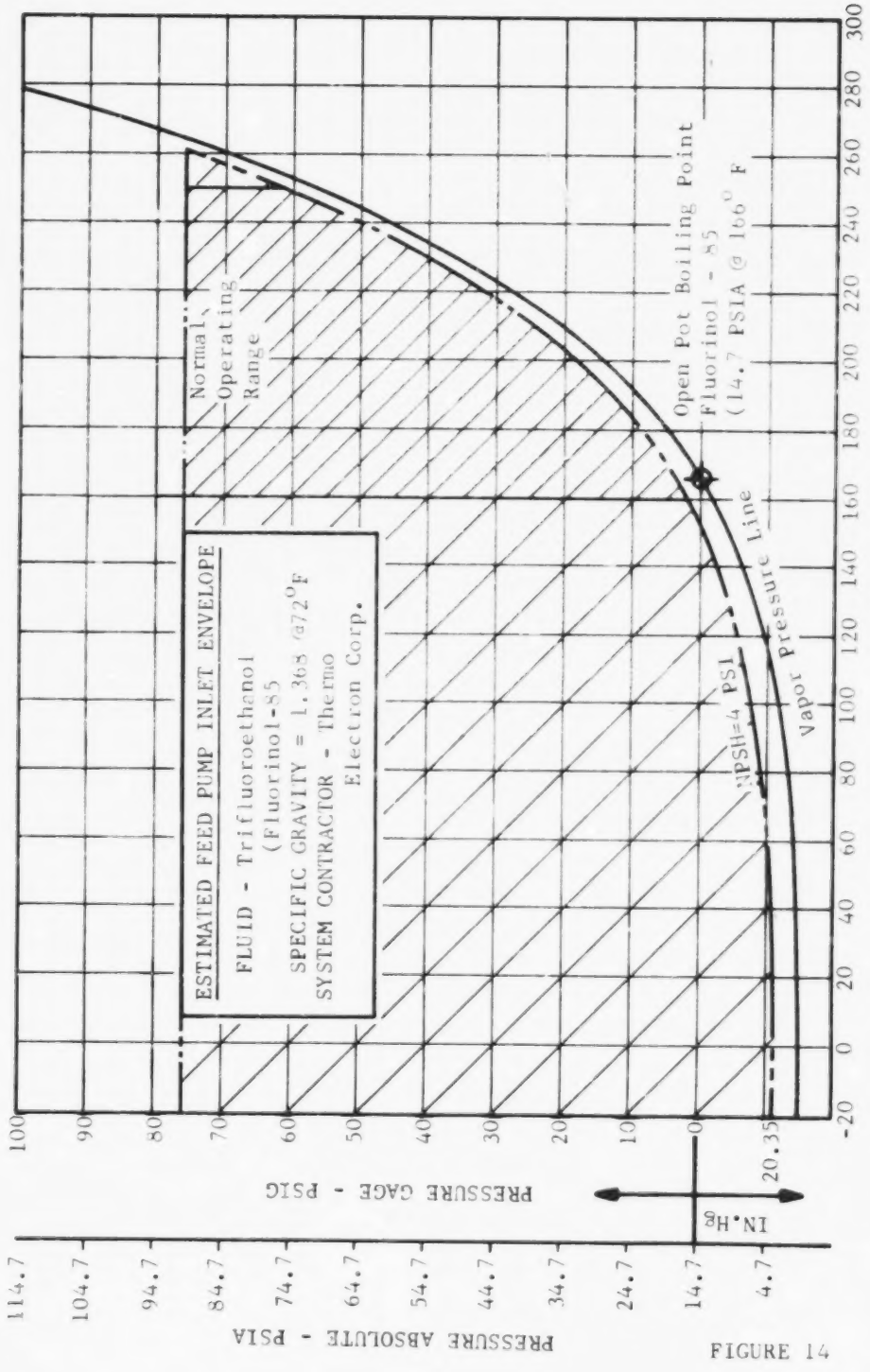

ESTIMATED FEED PUMP INLET ENVELOPE
FLUID - Trifluoroethanol
(Fluorinol-85
SPECIFIC GRAVITY = 1.363 @72°F
SYSTEM CONTRACTOR - Thermo
Electron Corp.
Normal Operating Range
Open Pot Boiling Point
Fluorinol - 85
(14.7 PSIA @ 166° F
NPSH=4 PSI
Vapor Pressure Line
TEMPERATURE - °F
-20 0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300
PRESSURE GAGE - PSIG
100 90 80 70 60 50 40 30 20 10 0
20.35
IN. Hg
PRESSURE ABSOLUTE - PSIA
114.7 104.7 94.7 84.7 74.7 64.7 54.7 44.7 34.7 24.7 14.7 4.7

FIGURE 14

THERMO ELECTRON CORPORATION

PUMP DRIVE AND CONTROL

TECHNICAL REQUIREMENTS

1. Drive Mechanism The feedpump shall be mechanically driven with power taken from the expander shaft or gearbox. The feedpump shall provide the total range of required flow and pressure with the expander operation at any speed within its specified speed range. The non-linear relationship between expander speed and required feedpump flow and pressure shall be accommodated by variable drive mechanism characteristics and/or variable feedpump capacity characteristics. The feedpump and drive mechanism should be capable of being mounted in the powerplant as required by the system contractor.

2. Control Movement of the accelerator pedal will provide a mechanical input signal to the feedpump for flow modulation.

TECHNICAL APPROACH

An outline drawing of the proposed Lear Motors pump is shown on the following page. The pump has variable displacement capability and can be directly driven by the expander. Pump to expander speed ratio and interface are not firm and will have to be worked out between Lear Motors and the system contractor.

Pump physical size is based on the contracts pump performance requirements, i.e. flow rate, and any change in the design flow rate will directly affect pump physical size and weight.

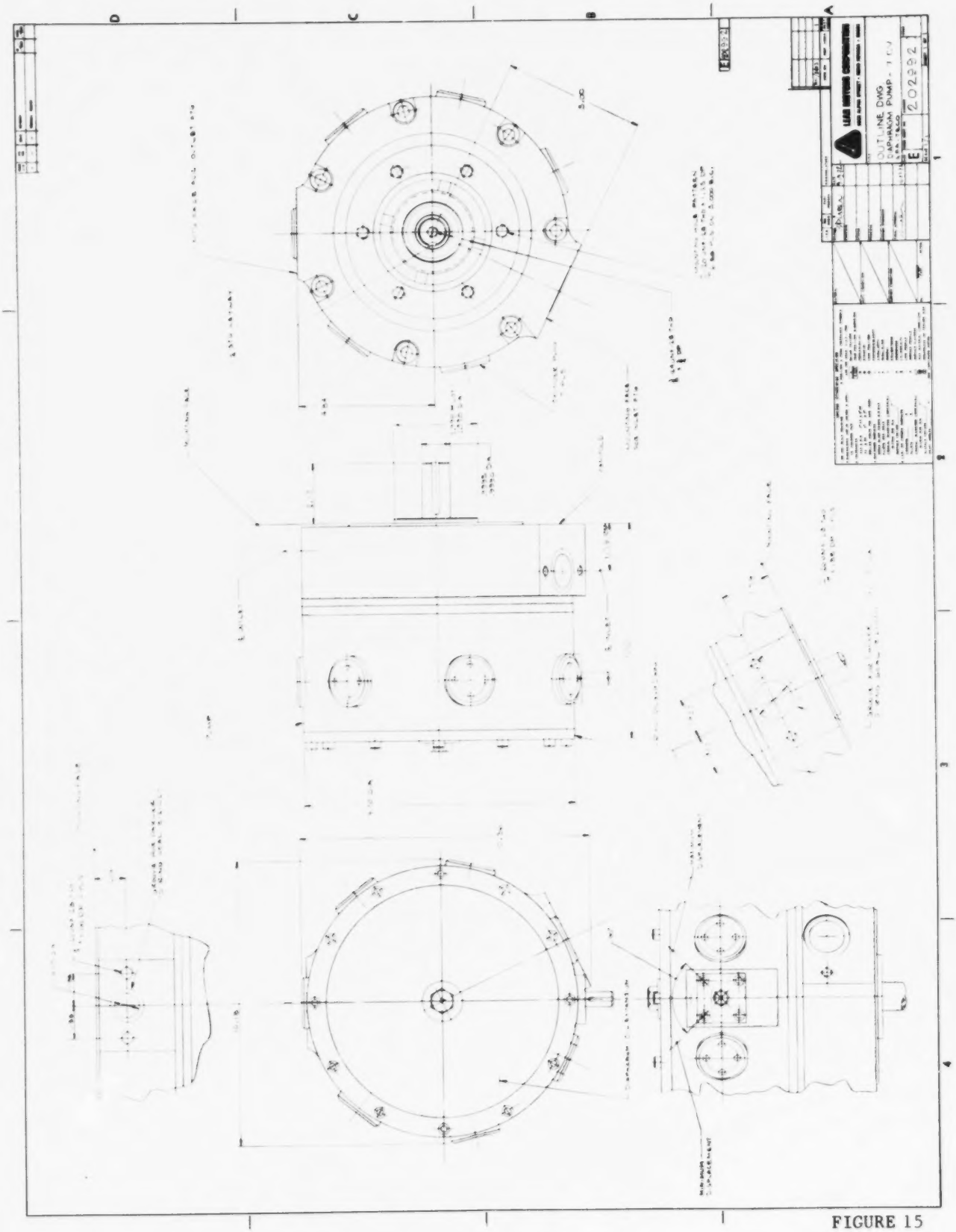


FIGURE 15

THERMO ELECTRON CORPORATION

GENERAL REQUIREMENTS

TECHNICAL REQUIREMENTS

1. Vehicle Design Goals Where applicable "Vehicle Design Goals-Six Passenger Automobile", shall be used as a guide for the design requirements.

2. Leakage There shall be no external leakage from the feedpump and its drive mechanism. The unit shall be capable of being hermetically sealed.

3. Lubrication Any lubricants that may be used within the feedpump and its drive mechanism shall not mix with or contaminate the working fluid. The lubricant shall be the same as that used in the expander.

4. Materials Compatibility All materials utilized in the construction of the feedpump shall not corrode or otherwise chemically react with the fluid in any manner.

TECHNICAL APPROACH

The Lear Motors pump design with its flexible diaphragm seal provides inherent hermetic sealing of working fluid from the air and lubricated section of the pump.

Materials used for the construction of the proposed TECO feedpump are shown in Table 3. The material used for the pumps flexible sealing member can be constructed from most elastomers which meet the temperature requirements and is compatible with the working fluid.

V CONCLUSION

Based on the results of this study and the past experience of Lear Motors in the development of Rankine Cycle Automotive Propulsion Systems, a radial piston, variable displacement feedpump design was selected to satisfy the requirements of the three Rankine Cycle System Contractors.

The proposed feedpumps hermetically seal the working fluid from the atmosphere and lubricating oil by utilization of a diaphragm seal. all of the highly stressed moving parts in the pump are oil lubricated allowing conventional, low cost materials to be used throughout its construction.

Three separate pumps were determined necessary because of the vast difference in system flow requirements. Due to the high flow at low speed requirements of the two organic systems, these pumps will be physically larger than the pump designed for the water base contractor.

Ductile iron was selected as the material of construction for the pump housing. A detailed stress analysis of an aluminum pump housing as a means of reducing weight is recommended. Also, any reduction in the high flow - low speed requirement of the organic systems would allow a significant reduction in size and weight.

Projected feedpump performance will contribute to high cycle efficiency; full displacement mechanical efficiencies from 85 to 90 percent are feasible.

It is recommended that an externally driven boost pump be used with the proposed feedpumps to assure reliable operation at all system operating conditions. A centrifugal type boost pump was selected because of its capability for pumping at very low NPSH values.

A major effort was directed toward finding elastimers which were compatible with the two organic fluids. Several candidates were found acceptable for operation at 250° or lower. Additional investigations are required to determine the long term effects of the fluids on elastomers.

The Lear Motors proposed feedpumps will satisfy the requirements of the three Rankine Cycle Automobile Power Systems with one exception;

the low flow - high speed region of the estimated pump performances does not fully meet the system contractors mechanical efficiency requirements.

The knowledge gained from this study and the background stemming from our four year feedpump development program are incorporated in these proposed pump designs. Reliable operation with a minimum development time is assured as this basic type of pump has been well developed in the Lear Motor Rankine Cycle Systems.

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APPENDIX A

LEAR MOTORS PUMP STANDARDS

7001-3401
July 26, 1972

PUMP STANDARDS
LEAR MOTORS CORPORATION

PUMP TEST CODE

I. Object

This code establishes a procedure for conducting and reporting tests of Rankine Cycle feedwater and condensate pumps. It is intended that the tests shall be made and reported by qualified personnel trained in the proper application and use of the various instruments and methods involved.

II. Records

Complete records shall be kept of all information relevant to a test. The serial number, type, size, or other means of identification of each pump tested shall be recorded in order that mistakes in identity be avoided.

III. Measurements

The essential measurements for test of feedwater and condensate pumps are:

1. Flow
2. Outlet Pressure
3. Suction Pressure or Vacuum
4. Temperature of pumped fluid
5. Temperature of pump
6. Power input to pump
7. Speed

IV. Flow Measurement

The rate of flow shall be expressed in gallons per minute (gpm) and be measured with calibrated rotameter compatible with pumped fluid.

V. Outlet Pressure Measurement

Head or outlet pressure shall be designated in pounds per square inch gage (psig). Measurement will be made with a calibrate pressure gage of the proper range.

VI. Suction Pressure or Vacuum Measurement

Suction pressure shall be designated in pounds per square inch absolute (psia). Suction vacuum conditions shall be designated in inches of mercury (in. Hg). Measurement will be made with a calibrated pressure gage or mercury manometer of the proper range.

VII. Temperature Measurements

Temperature will be expressed in degrees fahrenheit (°F). A calibrated thermometer with the proper range will be used.

VIII. Input Power Measurements

(a) Pump input horsepower may be determined by means of transmission dynamometers, torsion dynamometers or calibrated drivers.

(b) Transmission Dynamometers. When pump input horsepower is to be determined by transmission dynamometers, the unloaded and unlocked dynamometer must be properly balanced prior to the test at the same speed at which the test is to be run and the scales should be checked against standard weights. After the test, the balance must be rechecked to assure that no change

(b) Cont.
has taken place. In the event of an appreciable change, the test shall be rerun. An accurate measurement of speed is essential.

(c) Torsion Dynamometers. When pump input horsepower is to be determined by torsion dynamometers, the unloaded dynamometer shall be statically calibrated prior to the test by measuring the angular deflection for a given torque, the tare reading on the dynamometer scale being taken at rated speed with the pump disconnected. After the test, the calibrations must be rechecked to assure that no change has taken place. In the event of an appreciable change, the test shall be rerun. An accurate measurement of speed is essential.

(d) Calibrated Drivers. When pump input horsepower is to be determined by the use of a calibrated motor, the following shall be applied:

1. All measurements of power input shall be made at the terminals of the motor to exclude any line losses that may occur between the switchboard and the driver itself. Certified calibration curves of the motor must be obtained. The calibration shall be conducted on the specific motor in question, and not on an identical machine.

2. Such calibration curves must indicate the true input-output value of motor efficiency and not some conventional method of determinining an arbitrary efficiency.

3. When the pump is arranged with a speed changing device between the driver and the pump, the input to the pump shall be the actual output of the driving element less the loss through the speed changing device. The value of this loss shall be certified by the manufacturer of the speed changing device.

X. Measurement of Speed

(a) Measurement of speed shall be made by means of revolution counters or tachometers.

(b) For speed measurements taken by means of a revolution counter, the timing period shall be of sufficient length to obtain a true average speed and the stopwatch used should be checked against a standard timer.

(c) When a tachometer is used, it shall be calibrated against a revolution counter before and after test. Tachometer readings should be made at frequent intervals during each test point to obtain an accurate measurement of average speed over the reading period.

PUMP RATING CRITERIA

I. Introduction

The following numbered paragraphs define the quantities used to designate pumping applications.

II. Volume, Standard Units

(a) The standard unit of volume shall be the United States gallon or the cubic foot. The standard U.S. gallon contains 231.0 cubic inches. One cubic foot equals 7.4805 gallons.

(b) The rate of flow shall be expressed in gallons per minute (gpm), gallons per hour (gph).

(c) The specific weight of water at a temperature of 68°F shall be taken as 62.3 lb per cu ft. For other temperatures, proper specific weight corrections should be made.

III. Pump Volumetric Efficiency (Symbol E_v)

The volumetric efficiency of a rotary pump is the ratio of the actual pump capacity to the displacement, expressed in percent, at the specified pumping conditions:

$$E_v = \frac{\text{capacity}}{\text{displacement}} \times 100$$

IV. Pump Input (Symbol bhp or ehp)

The unit of power input to pump is the horsepower.

1 horsepower = 550 foot-pounds per second
= 33,000 foot-pounds per minute
= 2545 Btu per hour
= .7457 kilowatts

(a) The input horsepower (bhp) when measured by a transmission or torsion dynamometer, is calculated from the following formula:

$$bhp = \frac{2\ \pi LWN}{33{,}000}$$

where

L = length of lever arm in feet

W = net weight in pounds

N = speed in rpm

π = 3.1416

(b) The electrical horsepower input to an electric motor is given by:

$$ehp = \frac{kw}{.746} \text{ or } \frac{\text{Volts X Amps}}{746}$$

where

kw = kilowatt input.

(c) The input horsepower to a pump driven by an electric motor is:

$$bhp = ehp \times E_m$$

where

E_m = true efficiency of motor.

V. Pump Output (Symbol whp)

(a) Pump output is the liquid horsepower delivered by the pump.

(b) The liquid horsepower is to be computed by the following formula:

$$whp = \frac{\text{pounds of liquid pumped per min.}}{33{,}000} \times \text{Total head in ft of liquid}$$

(c) If the capacity is expressed in gallons per minute (gpm) the formula for liquid horsepower becomes:

$$whp = \frac{gpm \times (\text{total head in feet}) \times \text{sp gr}}{3960}$$

where:

sp gr = specific gravity of liquid referred to 68°F water, weighing 62.3 lb per cu ft.

(d) If the total head is expressed in pounds per square inch, the formula for liquid horsepower, irrespective of specific gravity of the liquid, becomes:

$$whp = \frac{gpm \times (\text{total head in psi})}{1714}$$

VI. Mechanical Efficiency

(a) Pump mechanical efficiency (E_p) is the ratio of the energy delivered by the pump to the energy supplied to the pump shaft; that is, the ratio of the liquid horsepower to the brake horsepower expressed in percent:

$$E_p = \frac{whp}{bhp} \times 100 \quad \text{or} \quad \frac{whp}{ehp \times E_m}$$

(b) Overall unit efficiency (E_o) is the ratio of the energy delivered by the pump to the energy supplied to the input side of the pump driver; that is, in the case of electric-driven pumps, the ratio of the liquid horsepower to the electrical horsepower input to the driver, expressed in percent:

$$E_o = \frac{whp}{ehp} \times 100$$

VII. NPSH Definition

NPSH (net positive suction head) is defined as the net pressure absolute above vapor pressure at the pump inlet. In pumping liquids we are concerned with getting the liquid into the pump in a liquid state (i.e. without vaporization). Suction lift, friction losses, and entrance losses all conspire to reduce net pressure and in some cases interfere with the liquid getting into the pump in liquid phase.

It should be remembered that the boiling point of a liquid is that point at which vapor pressure equals external pressure. Whether this takes place in open or closed vessels there is usually a vapor area in contact with the liquid and a liquid level line. At boiling point the addition of heat or the lowering of external pressure unbalances this equilibrium and results in the vaporization of an amount of liquid.

Also to be considered is that centrifugal pumps are liquid handling machines. Although pumps will handle up to 50% entrained gas or vapor and will dispose of the air in suction lines during priming cycle they are not efficient vapor handli g devices. Further, there is a great expansion of volume in the conversion from liquid to vapor state which takes up the inherent pump capacity.

Reduced capacity at low NPSH is an efficiency loss permitting no revision of brake horsepower. Because of this major consideration of maintaining liquid state during the pumping cycle it is essential to make an NPSH calculation in all applications involving a liquid at or close to boiling point.

NPSH CALCULATION

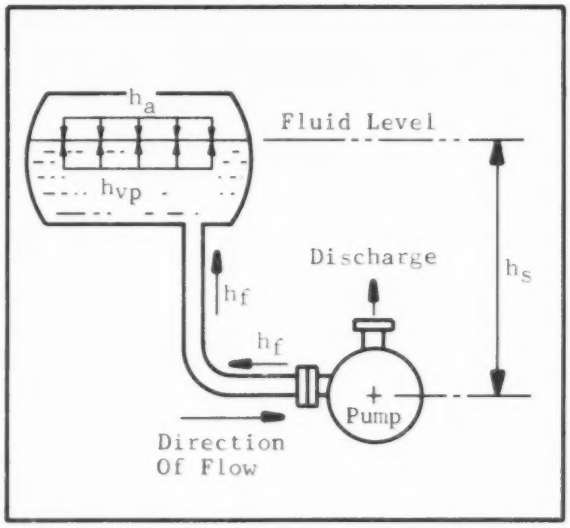
Three general cases may be considered in making an NPSH calculation.

Case I - The liquid level of the supply is above the pump centerline. This is termed static suction head.

Case II- The liquid level of the supply is below the pump centerline. This is suction lift.

Case I is by far the most common in industrial applications. Hot water or various other liquids are pumped out of a vessel elevated above the pump. In such a case it is usually necessary to consider only four elements in calculating NPSH.

CASE I



WHERE:

h_s = Static Suction Head (PSIG)
h_{vp} = Vapor Pressure (PSIA)
h_a = Atmospheric Pressure (PSIA)
h_f = Friction Loss (PSIG)

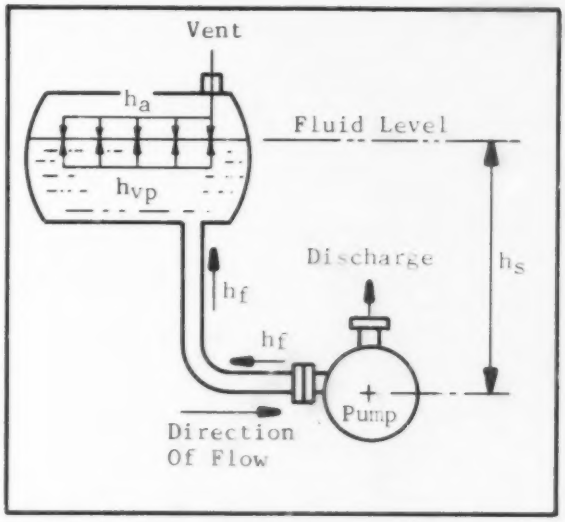
In the range of capacities involving pumps (up to 250 GPM) velocity head need not be considered.

Then in any Case I system the following formula applies:

$$h_s + h_a - (h_{vp} + h_f) = NPSH$$

Case IA: The liquid is in a closed vessel in vacuum or pressure systems. In this special case vapor pressure will equal atmospheric pressure.

CASE IA



WHERE:

h_s = Static Suction Head (PSIG)
h_{vp} = Vapor Pressure (PSIA)
h_a = Atmospheric Pressure (PSIA)
h_f = Friction Loss (PSIG)

$$\therefore h_s - h_f = NPSH$$

In planning piping for NPSH problems it is usually desirable to increase the size of suction piping until friction loss in the suction line is less than 6 in. of liquid head.

In such cases h_f may be disregarded in the application. The formulae then becomes:

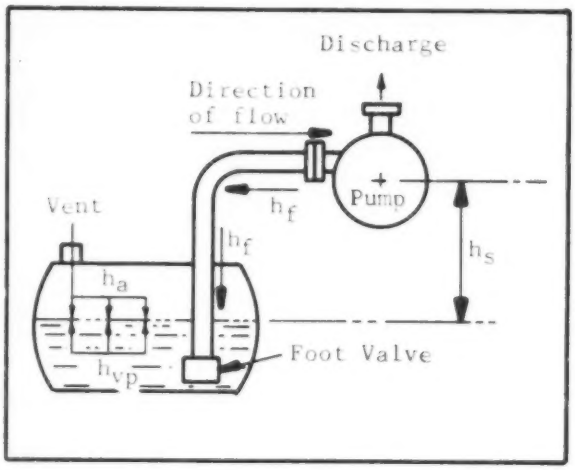
Case I: $$h_s + h_a - h_{vp} = NPSH$$

Case IA: $$h_s = NPSH$$

In Case I NPSH is identical with static suction head when the liquid is at boiling point and the suction piping is large enough to keep total friction loss below 6 in. of liquid head.

Case II can never involve liquids at boiling point since a negative NPSH condition would result and vaporization would take place. All cases of NPSH calculations involving suction lift must also involve liquids below boiling point. With the vapor pressure less than the atmospheric pressure the formula is as follows:

CASE II



WHERE:

h_s = Static Suction Head (PSIG)
h_{vp} = Vapor Pressure (PSIA)
h_a = Atmospheric Pressure (PSIA)
h_f = Friction Loss (PSIG)

$$h_a - h_{vp} - h_s = \text{NPSH}$$

Atmospheric pressure is then the only positive force in Case II. Both vapor pressure and suction lift are negative factors.

NOTE:

Static suction head (h_s) for any fluid:
Inches of Head x Specific Gravity x .03613 = PSIG
Feet of Head x Specific Gravity x .4335 = PSIG

Max K. Winkler

Max K. Winkler

Reference: Standard of Hydraulic Inst.